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DESIGN · PRODUCTION · MATERIALS

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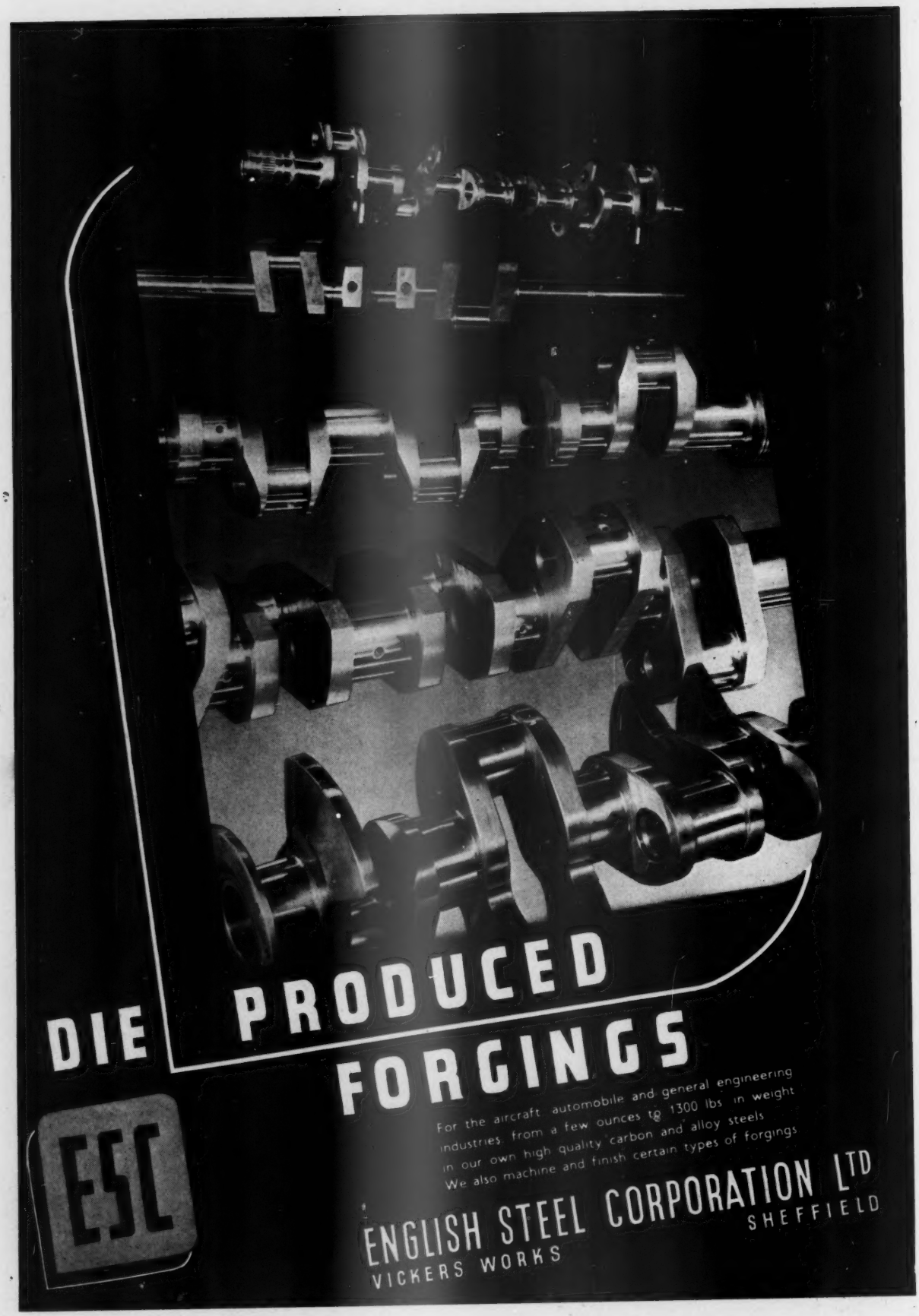
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Design, Materials, Production Methods, and Works Equipment

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M. I. R. A.

THE Association's annual report for 1945/50 is in many senses a satisfactory document. Not only is the Association able to report a sound financial position, but what is even more important, a more active interest by the industry in the work of the Association and its services. It is for example, reported that there is considerable increase in the library loans, which now amount to a considerable number. There are also an increasing number of visitors seeking advice and assistance.

Good progress has been made in many fields, notably the subject of electronics and instrumentation. It is stated that the Association finds itself frequently consulted ; they are in fact looked to for expert advice on the subject of electronics, not only by Members, but by official bodies. It is believed that the Association is now generally regarded as an authority in this field.

Perhaps the highlight of the years' activities is the considerable expansion that has taken place in the experimental facilities at the Association's outdoor laboratory or proving ground near Nuneaton. The stretch of pavé road having been completed, it now seems to be one of the most widely used of the various facilities that are provided. Other experimental road surfaces already constructed or nearing completion include an example of a typical overseas "washboard" or corrugated road. There are also tracks with surfaces designed to assist in the investigation of tyre and road noise. There is also a typical example of a cross-country track.

The Dust Tunnel

Another valuable facility in view of the export drive is the experimental dust tunnel. A water splash has also been arranged. In this connection and in relation to the whole aspect of vehicle sealing, it is to be hoped that facilities will ultimately be available for the investigation of external air flow conditions. At times, the best of sealing appears to break down, and this is possibly due to some particular prevailing air condition at the areas evidencing the weakness. A sustained high pressure blast will carry in the finer dusts, and also rain and storm water, if such pressures are sufficiently heavy and long sustained.

It is especially satisfactory to note that the Association appear to be more satisfied now with the industry's response to their activities. In other words, there appears to be a measure of co-operation and a definitely growing tendency to make full use of the facilities that the Association was created to provide. There is in fact continual demand for more and more facilities. This growing interest and development of course must necessarily raise the whole question of finance, and it is understood that while this is at present satisfactory, increases must be looked for if the work and growth is to continue satisfactorily. Valuable assistance has been received from the Department of Scientific and Industrial Research. The overall finance still of course remains at a very low figure indeed, in proportion to the total financial turnover of the industry. It is stated that income and expenditure are about equal at approximately £70,000 per annum. A valuable contribution from the Society of Motor Manufacturers and Traders is reported in the form of a special grant of £10,000.

The New Laboratory

As regards the new laboratory, progress has been slower than was hoped, owing largely to certain especial difficulties in connection with the land that had been selected. A number of difficulties had to be surmounted, particularly as this site has a county boundary running through it. Final permission has however now been obtained, and it is hoped that the scheme will rapidly go ahead. The site of the new laboratory is about a quarter of a mile from the proving ground, a new road will be made.

The laboratory will have approximately double the floor space of the Brentford premises, and as the site is one of 11 acres, there will be plenty of land for future extension. It is intended that there should be a canteen for the Association staff, with catering and sleeping accommodation for the staff of member firms using the proving ground. There will also, of course, be a building for garaging vehicles used on the proving ground.

Regarding the proving ground, contracts have been placed for extensive re-surfacing and for the construction of a steering or skid pan. Other essential facilities also are being investigated by the proving ground Sub-Committee. A major project in this connection is the provision of a high-speed circuit. A detailed scheme has been prepared for a track suitable for sustained speeds of 100 miles per hour. This scheme, which has been approved by the Council, involves very considerable expenditure, and the cost would be more than the Association could meet from present resources. Consultations have therefore been initiated with the S.M.M.T. and the D.S.I.R., to raise the necessary funds.

Future Tendencies

Much interest attaches to the remarkable design project sponsored by General Motors Styling Division, representing a forecast of the automobile of 1960. Although full mechanical details are not available, it is evident that the car embodies a number of features that are even now accepted as desirable and indeed proven in practice by contemporary European engineers. The significance of the project is perhaps all the more marked on this account, since so many attempts at forecasting the future have been made either with a complete disregard of mechanical design or with a faith in the rate of technical development that could not be justified on the basis of past experience.

In other words, the General Motors project is firmly based on design features that are known to be sound in that many of them have already been proved on some of the more highly developed European vehicles. In particular, the use of De Dion rear suspension would seem to confirm the view held in many quarters that it represents the best practical solution to the problem of rear suspension that can be put forward in the present state of the art. Eventually, no doubt, fully independent suspension will come, but since the formidable mechanical problems associated with the use of soft suspension at the front in conjunction with the beam axle do not apply to the rear axle, it is unlikely that concentrated effort will be directed to this development. There are problems of greater immediate concern that require solution.

Integral Drive

It is interesting to observe that the transmission and final drive form an integral unit and that the brake drums are adjacent to the differential housing. This again reflects a tendency evident in the more advanced European designs, one example of which is described in this issue of the *Automobile Engineer*. Although the total weight of the car is expected to be about 3,000 lb., the proportion of this weight carried on the front and rear wheels is not given. From the layout of the car, however, it appears that the transfer of a considerable proportion of the power unit and transmission mass to the rear end is necessary from the point of view of weight distribution and handling. Further, the location of 40 gallons of fuel in tanks installed in the wing fairings behind the rear wheels points to the same conclusion.

Since the engine output is expected to exceed 300 h.p., the car is obviously capable of extremely high speeds and in these circumstances a high polar moment of inertia seems to be desirable. In general layout the front suspension is orthodox except for the use of rubber in torsion as the springing medium. The retention of what has come to be regarded as the conventional wishbone arrangement suggests that as regards front suspension, opinion has finally hardened on the question of the most practical and satisfactory compromise. This is perhaps not surprising, since more thought and effort has possibly been concentrated on front suspension in the last fifteen years than on any other aspect of chassis design. It is not, therefore, unreasonable to suggest that finality is being approached in this matter.

Regarding engine layout it is to be expected that the 90 deg. V-eight should be adopted. In making a forecast of the shape of cars in ten years' time, this layout could hardly have been otherwise, for with the introduction of so many new engines of this type in the last two years, it is inevitable that from production considerations alone the American industry is committed to the V-eight engine for this period at least. Further, it is also inevitable that something approaching the "square" engine must be adopted, and this unit with a displacement of 215 cu. in. has a bore and stroke of $3\frac{1}{4}$in.

The Supercharger

All this seems to be a sound and logical foundation for the design, but the introduction of a supercharger is a more open question, not only on account of fuel consumption considerations, but also because of the evident need to employ methanol as a fuel under conditions when the engine is giving a high power output. At the moment, with a 10 to 1 compression ratio, premium fuels are required for normal operation, and methanol at the higher b.m.e.p.s. Whether in ten years' time the quality of premium grade fuels will permit the use of a single fuel for all running conditions, is a matter that is debatable, so say the least, and the advisability of placing the responsibility on the public of buying two fuels seems open to question.

This may appear a trifling point, but greater projects have failed for even less significant reasons. In a country where the car buyers are primarily concerned with transport and not with ministering to the needs of an elaborate mechanism, the car owner might justifiably resent the extra trouble, small though it is, incurred every time that he visits a filling station. This, however, may be looking too far ahead and from the point of view of a development project the inclusion of a supercharger in the specification may well be justified.

Of the many elaborate coachwork details little need be said. It is unlikely that all of them could find a place on a production vehicle to be sold at an economic price, but the accumulated experience will be of inestimable value in sorting the grain from the chaff. If a word about the general styling of the car may not be out of place, attention should once more be drawn to the obsession of the stylist in trying to make a car appear as if it were an aircraft, or for that matter any other vehicle but a car. This surely cannot be regarded as a serious and lasting form, and should be dismissed as soon as possible for what it is, namely a manifestation of immaturity of thought.

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THE LANCIA "AURELIA"

A Distinctive Italian Design, Embodying Many Novel Features

THE makers of the Lancia "Aurelia" chassis are noted for their consistent and successful departures from the orthodox. Twenty-seven years ago they produced the outstanding Lambda chassis, embodying independent front suspension with vertically sliding steering swivels and integral shock absorbers. The original Lambda engine was unusual in that the four cylinders were arranged in a close-set Vee, so that the overall length of the power unit was much reduced. The same scheme was followed on the larger vehicle also, the engine being a Vee 8, also at 24 deg. In the latest car, the Vee power unit is continued but in this instance, the cylinders, six in number, are in two banks of three at 60 deg. The six-throw crankshaft has four-bearings with adjacent crankpins at 60 deg.

A power unit some 6½ inches shorter than would otherwise be the case is the result of this arrangement, while the space it occupies is still further reduced by the removal of the clutch to the rear of the vehicle. Here, it is incorporated with the combined gearbox and differential assembly, rubber-mounted on the frame and driving the independently-sprung rear wheels through universal shafts. To provide adequate length for the shafts the outer universal joints are outside the rear wheel hubs.

Integral body and chassis construction is adopted, a notable point being the absence of centre door pillars. Six passengers can be carried and the complete car weighs 22¼ cwt. Like all Italian cars, the "Aurelia" is intended for sustained high speeds and the 1,754 c.c. engine is so geared that the peak of its power curve at 4,000 r.p.m. corresponds to a road speed of 76 miles per hour. The front suspension is fairly stiff, the wheel displacement being in any case definitely limited by the design of the suspension. The rear suspension is softer, but both are firmly damped by hydraulic shock absorbers to ensure road holding and good handling at speed, rather than to provide maximum comfort at moderate speeds.

SPECIFICATION

ENGINE. Six cylinder, 60 deg. V, 70 mm. bore, 76 mm. stroke, capacity 1,754 c.c. Inclined overhead valves in hemispherical heads, push-rod operated. Compression-ratio 6.85 to 1. Maximum b.h.p. 55.2 at 4,000 r.p.m. Maximum b.m.e.p. 110 lb. per square inch at 2,750 r.p.m.

TRANSMISSION. Rubber - jointed, divided propeller-shaft connecting to 6⅝in. diameter, single-disc clutch on gearbox. Four speeds (all indirect), Warner baulked synchromesh on top, 3rd and 2nd speed. Spiral bevel final drive combined with gearbox, universal shafts to independently-sprung rear wheels on diagonally-swinging arms.

Gear ratios: Top 4.05 to 1 : 3rd 5.83 to 1 : 2nd 8.66 to 1 : 1st 13.4 to 1. Reverse 17 to 1.

FRONT SUSPENSION. Vertically-sliding I.F.S. with integral hydraulic shock absorbers. Worm and gear steering, with track rod and cross drag link.

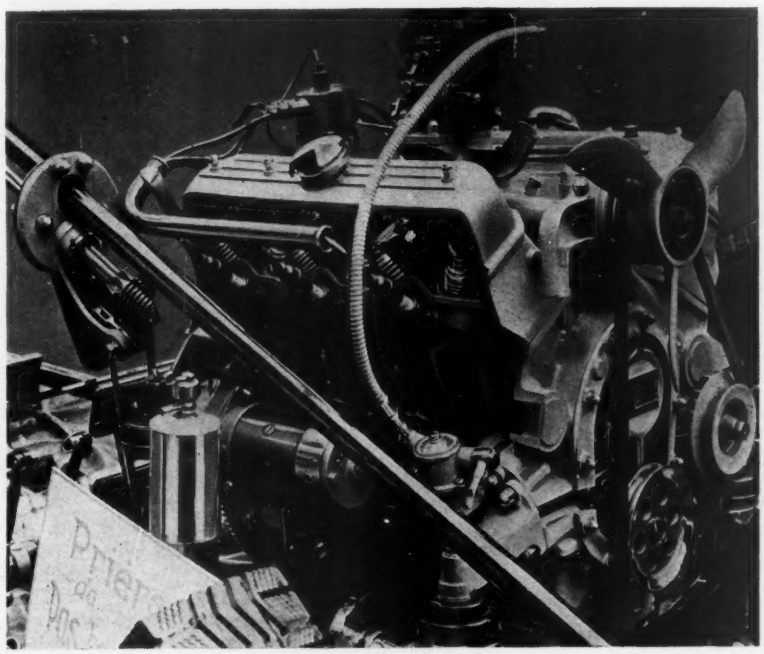
BRAKES. Hydraulic, floating cam, drums 275 mm. dia., 60 mm. wide. Cable-operated parking brake.

Engine

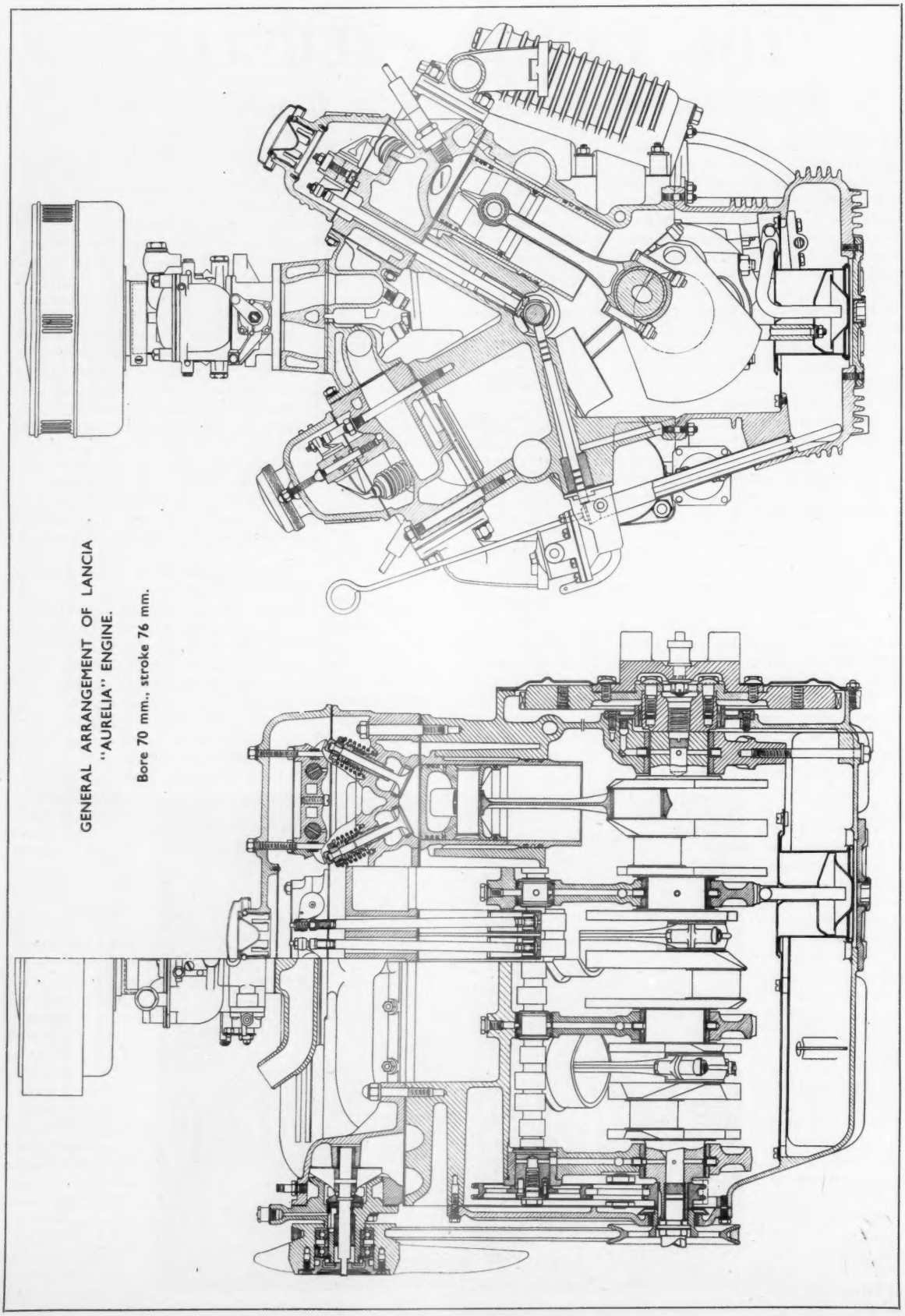
With a bore of 70 mm. and a stroke of 76 mm. the capacity is 1,754 c.c. The compression-ratio is 6.85 to 1 and at 4,000 r.p.m. a maximum b.h.p. of 55.2 is obtained, while maximum b.m.e.p. is 110 lb. per square inch at 2,750 r.p.m. Although the engine has overhead valves in hemispherical combustion chambers, the power output is not as great as might be expected from a type of motor generally associated with high mean effective pressures at high speeds. In any case the car is so light and its windage so low that the power actually available is ample and will give sustained road speeds of over 80 miles an hour. On account of the high top gear ratio, third speed is in any case required for real acceleration and high engine torque is not therefore so necessary.

Cast in aluminium alloy, the cylinder block comprises two banks of three bores each set at 60 deg. The left-hand bank is offset 44 mm. ahead of the right-hand one and one right- and one left-hand connecting-rod lie between adjacent bearings of the four-bearing crankshaft. The dimension between the extreme front and extreme rear cylinder bores is 288 mm. With a normal six-cylinder layout having say 20 mm. between adjacent bores to give the same coolant space as on the Aurelia, the overall centre distance would have been 450 mm., a saving of 162 mm.

Wet liners are fitted and just below the upper ends there are flanges that seat on faces machined on the upper ends of jackets completely enclosing each liner, leaving a small water space. This is fed by a large gallery running along the outside of each bank, the water escaping into the cylinder heads by passages cored in the supporting flanges of the liners.



Sectioned engine showing valve layout.

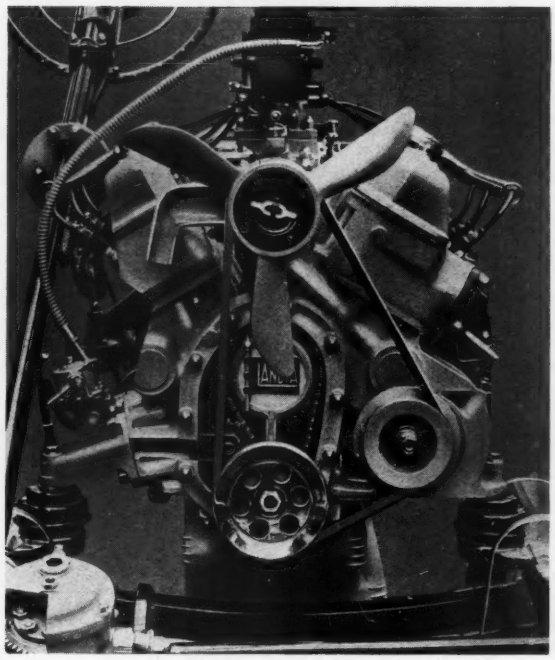


GENERAL ARRANGEMENT OF LANCIA "AURELIA" ENGINE.

Bore 70 mm., stroke 76 mm.

Near their lower ends the liners are machined to a push fit in the block, with three grooves, the upper and lower ones having synthetic rubber seals. The centre one is clear, and drained to atmosphere by a small drilled hole.

The sump joint is on the crankshaft centre-line and the deep, "I" section bearing caps are secured by two studs each. The white-metal lined steel bearing shells are 5 mm. in overall thickness and the third bearing from the front is flanged to give end-location to the crankshaft. The thick steel shells seem a highly commendable feature, as providing sufficient stiffness to hold the bearing bores more or less constant under extremes of temperature and thus reduce the tendency of crankshaft bearings to "freeze up" by the contraction of an aluminium alloy crankcase in cold weather.



Camshaft drive and front engine mounting.

Of simple rectangular form, with ribbed lateral enlargements in the lower portion, the aluminium alloy sump, with a thickness of 4.5 mm., makes an appreciable contribution to the stiffness of the cylinder block. A semicircular chamber is formed at the rear, which, co-operating with a similar one on the block, houses the thin flywheel and starter ring. A pressed steel cover completes the enclosure and has a central opening through which projects the die-cast aluminium alloy driving member of the rubber-block universal joint taking the drive to the clutch located in the rear-mounted gearbox.

Carried on four main bearings 54 mm. in diameter by 30 mm. wide the crankshaft has six crankpins, 45 mm. in diameter by 25 mm. They are spaced at 60 deg. in a continuous spiral, No. 6 crankpin coming back to 60 deg. of No. 1. The webs are machined all over and form integral balance weights. It will be understood that the spacing of the crankpins, in conjunction with the 60 deg. angle of the two banks of cylinders, gives equally spaced firing at 120 deg. as in a normal engine. Primary and secondary forces cancel out to the same extent as in an orthodox design. There are however appreciable primary and secondary rocking couples, though these are less than they would be if two cylinders worked on one crankpin. Judicious rubber mounting will probably have eliminated all unpleasant effects and the design retains two outstanding advantages of the six-cylinder layout. These are reasonably uniform explosion torque and absence of torsional reactions on the crankcase and transmission of the force required to accelerate and retard the reciprocating masses. In a four-cylinder engine these reach very high figures and are particularly noticeable on the overrun.

The steel connecting-rods are of normal design, though the white-metal lined big-end shells as previously noted, are thicker than usual at 3 mm. overall. The gudgeon-pins float in the aluminium alloy pistons which have three compression rings and one slotted scraper above the pin and a second slotted scraper in the skirt.

A double roller chain with a tensioning idler loaded by oil pressure drives the centrally-mounted case-hardened steel camshaft. This runs in split bronze bushes located by flanges on the shaft, held together for assembly purposes by circlips and prevented from turning by setscrews. The tappets are built up of a hardened end, pressed into a carbon steel tube which is secured by swaging its lower edge into a recess machined in the foot. The tappets run direct in the metal of the cylinder block and actuate the usual push-rods and rockers. Inclined at 26 deg. on either side of the vertical the valves lie all in the longitudinal central plane of each bank of cylinders, this being just possible by the wide spacing of the individual cylinders in each bank.

Each rocker consists of a central journal, carried in a split bearing block secured to the head, and diagonally-extending arms for the adjusting screw and valve-operating tip. The inlet valves are 26 mm. in diameter and the exhaust 24 mm. Double valve springs are employed and hardened caps are fitted to the valve stems.

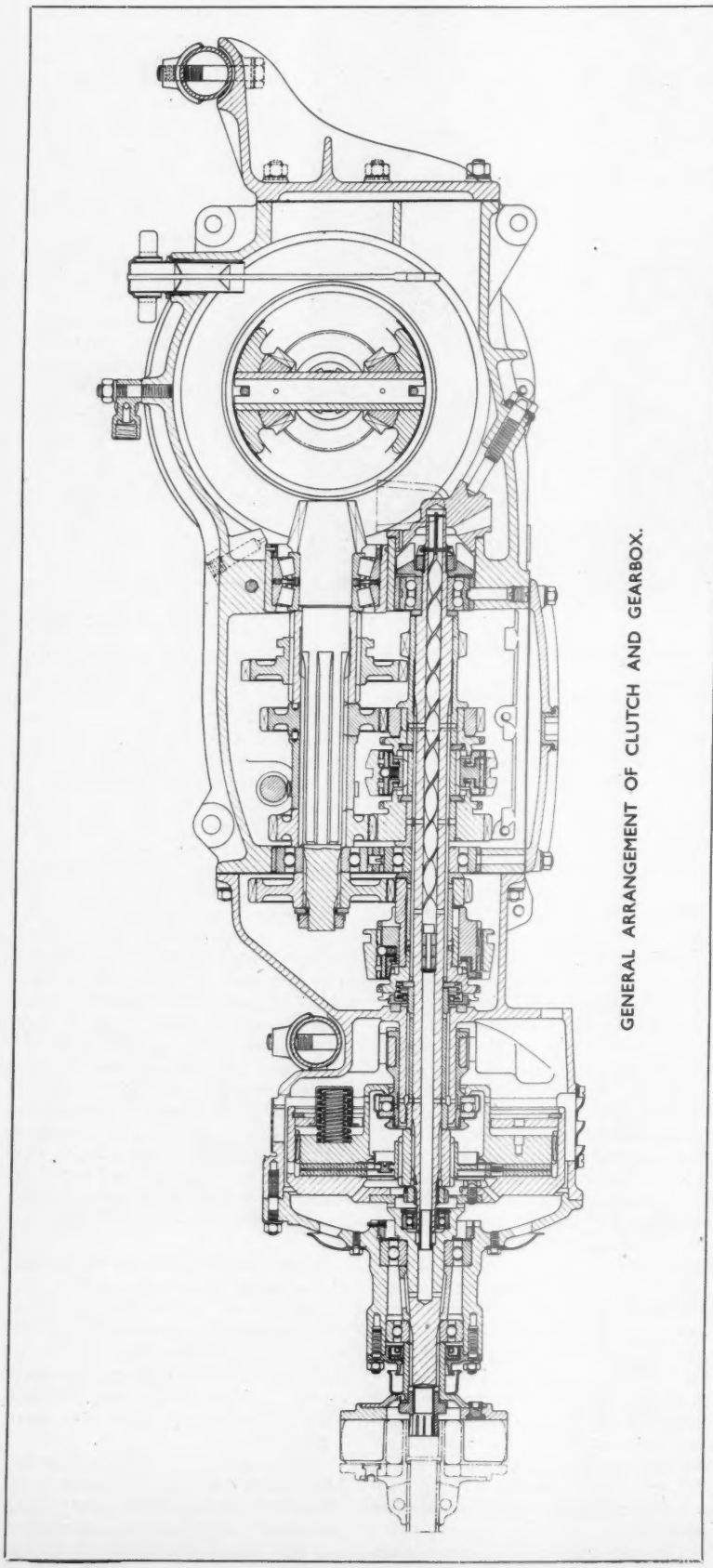
The 14 mm. sparking plugs are arranged conveniently at an angle of 27 deg. to the longitudinal plane. The whole layout is evidently an excellent solution of the problem of working all the valves in a hemispherically-headed engine off one camshaft while leaving the sparking plugs ideally accessible. Such a scheme is, of course, only possible on account of the staggered Vee layout which results in adjacent cylinders being 122 mm. apart. The cylinder heads are of aluminium alloy with inserted valve seats, and the rockers are enclosed in cast alloy covers. With a view to reducing resonant noises, the covers are 6 mm. thick, which is thicker than the figure for the block and heads.

Each bank of three cylinders has a water-jacketed manifold with three branches forming a cover to the whole side of the head water jacket. This communicates, by drilled passages, with the water jacket on the twin riser which connects the two manifolds with the dual downdraught carburetter. The two throttle spindles are mounted in a fore and aft direction but are rotated in opposite directions by means of spur gearing, thus avoiding any butterfly bias in the induction manifold.

The gear pump is located as usual at the bottom end of the slightly inclined distributor shaft, which is driven from a spiral gear on the rear of the camshaft. The pump suction is connected by a pipe to a strainer under No. 3 main bearing and carried by a distance piece extending down from the bearing cap.

A large circular cover in the bottom of the sump gives full access to the strainer gauze. A full-flow filter in an aluminium housing is bolted to the left side of the cylinder block and oil is delivered to a drilled gallery communicating by cross-holes with vertical drillings feeding the main and camshaft bearings.

Each crankpin is fed by a diagonal hole drilled through the chamfered tip of the web, through the crankpin up to crankshaft centreline, where it is fed by two radial cross-drillings in the



GENERAL ARRANGEMENT OF CLUTCH AND GEARBOX.

adjacent journal. The outer end of the diagonal hole, which is quite large for about half of its length and helps to lighten the crankpin, is closed by a plug. A hole drilled in the pin, at right angles to crankpin radius, feeds the big-end at what is now being recognised as a more strategic point than the radial position formerly common.

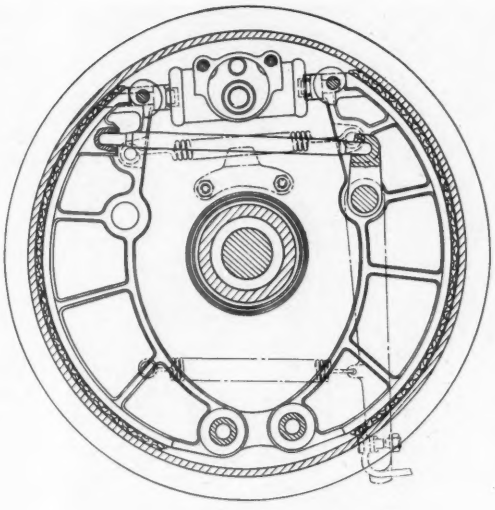
Triangulated with the dynamo, a vee belt drives the two-bladed cast aluminium fan, mounted on two closely-spaced ball bearings secured by a ring nut to a nose-piece integral with the pump cover. The pump spindle runs in plain bushes in the nose-piece and in the pump body at the rear. The flatted forward end of the pump spindle engages a slot in a cover plate attached to the fan hub and is also drawn forward by a wire spring passing through a hole in the projecting tongue. This spring gives a slight end-pressure on the spindle to keep the carbon-ring seal in contact with a washer pressed into the back face of the pump cover. The ring is driven by a cross-peg engaging keyways and sealed to the shaft by a rubber ring.

The pump body is studded to a facing on the cylinder block between the two cylinder heads and drives the water axially along passages cored in the block and feeding the individual cylinder jackets. Since pump delivery is downwards, a small rubber tube, connected to the radiator top tank, is necessary to prevent an air lock. There are two controls on water temperature, the usual thermostatic valve in the water outlet and in addition, thermostatically controlled shutters in front of the radiator.

Clutch

Secured to the rear face of the flywheel by flange and setscrews is a die-casting with four radial vanes 112 mm. in overall diameter and 27 mm. wide. These vanes, which end at an internal diameter of 54 mm. engage radial slots moulded in a rubber ring, four intermediate slots in which are engaged by similar vanes on a member attached to the front end of a divided coupling shaft with an intermediate bearing. An identical joint at the rear, both having sleeve and sphere centering spigots in addition, is attached to a flange splined to a short shaft carried on two ball bearings housed in the front cover of the clutch casing. The flanged rearward end of the stub shaft carries a small cast-iron flywheel, 187 mm. in diameter, enclosing a pressure plate engaged by a number of helical springs and withdrawn direct by a cross-lever operating a sleeve carrying a ball thrust bearing.

The clutch disc is only 168 mm. in diameter (about 6⅝in.) and is splined to a hub on the driving shaft of the indirect gearbox, the front end of which is supported by a ball spigot bearing in the stub-shaft flange. All the bearings get a very restricted supply of oil from the hollow driving shaft, which is fed with oil at a very slight pressure by a centrifugal pump discharging over a weir about 2in. high. Oil under this slight head percolates through a restriction plug made up of felt washers and reaches the various bearings by radial drillings. The same supply, but unrestricted by any plug, is flung outwards by radial holes to feed the idling bushes of the top, 3rd and 2nd speed gearwheels of the gearbox.



Section B-B

Arrangement of rear brake.

Gearbox

Housed in one casting with the differential driving gear, the gearbox is of the indirect type, with shaft centres of 75 mm. The driving shaft is below the driven, on the rear end of which, is the spiral bevel pinion for the final drive.

The bearings are mounted in steel housings located by grubscrews, and the driving shaft has a single-row ball bearing at the front end and at the rear. Where a small diameter only is possible, a double-row bearing is used, end location being by a grubscrew passing through the housing and into a groove machined in the outer race. The driven shaft has a ball bearing at the front end and opposed taper roller bearings at the bevel pinion end. The helical gears have face widths of 16 mm., the straight-toothed first-speed wheel is 14 mm. wide. Second speed is in front of the gearbox wall with top and third speed behind it.

As is general with indirect gearboxes the top speed is a slight overdrive at .862 to 1, which, with 10/47 pinion and crown wheel gives a ratio of 4.05 to 1.

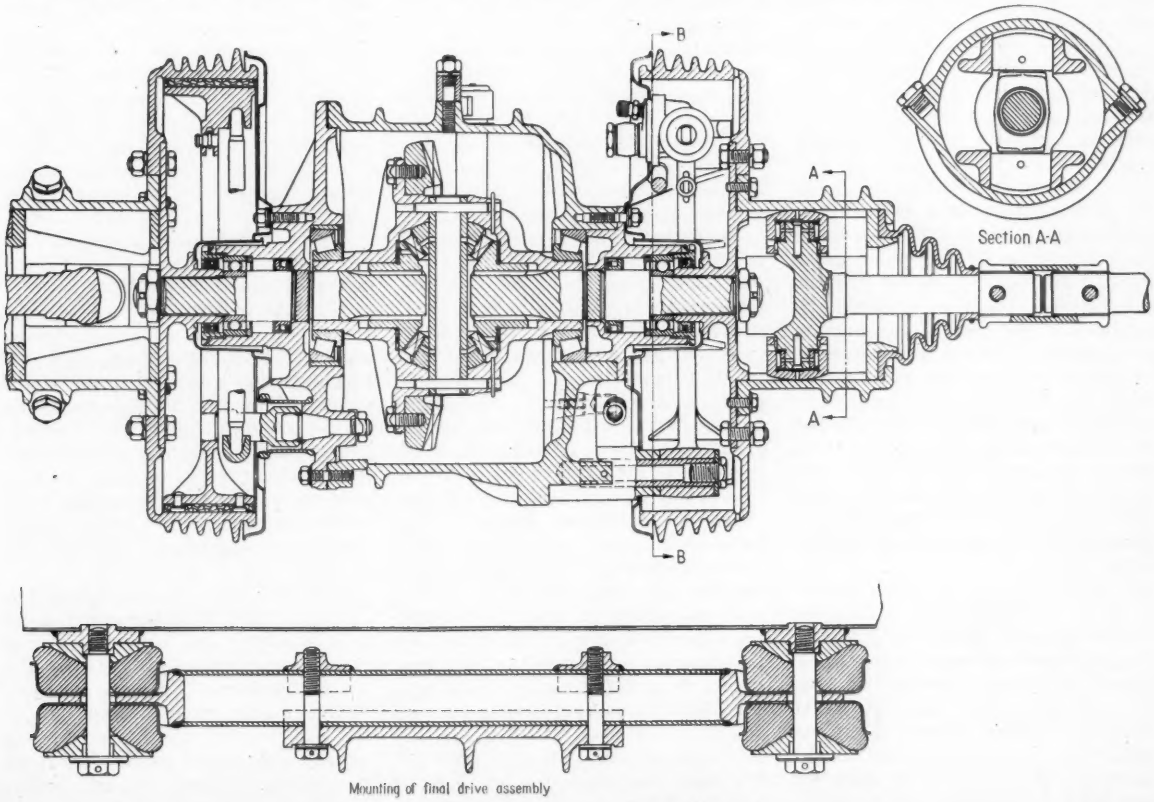
With top speed as unity, the gearbox ratios are :—

Top	1 to 1
3rd	1.44 to 1
2nd	2.26 to 1
1st	3.31 to 1
rev	4.2 to 1

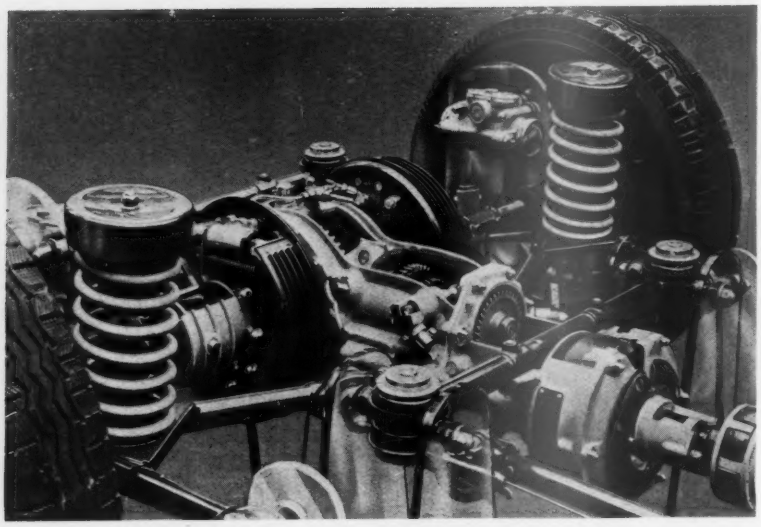
The overall ratios are :—

Top	4.05 to 1
3rd	5.82 to 1
2nd	8.65 to 1
1st	13.44 to 1
rev	17 to 1

With 165 by 400 Michelin tyres a road speed of 134 kilometres (83 miles) per hour corresponds to an engine speed of 4,500 r.p.m. The engine power curve is practically flat-topped at this point and shows 55 b.h.p. The windage of the car being low it settles down to 83-84 miles per hour on full throttle. Top gear hill-climbing is, as would be expected from



General arrangement of final drive.



Sectioned clutch, gearbox and final drive assembly.

an obviously high gear ratio, not remarkable, being about 6.4% with full load at maximum engine torque, corresponding to 74.5 kilometres (46 miles) per hour.

Warner baulked synchromesh is provided for the three higher speeds, the wheels of which all idle on steel sleeves on the driving shaft. The synchronising cones are, as usual with this type, on the small side at 52 mm. mean diameter. Since they are all on the driving shaft, all are equally effective in accelerating or retarding the spinning parts. Synchronisation of second speed, for example, is about twice as effective as in a normal gearbox.

The column-mounted change-speed moves in a longitudinal direction, by lever, ball jointed rod and bell-crank, a universally-jointed tube spanning the considerable distance to the gearbox. Selection of the striking forks is by rotation of the tube, the coupling mechanism consisting of a worm on the column tube meshing with a worm-wheel attached to a hanging lever giving fore-and-aft motion. The helical form of the worm is, of course, purely incidental to ease of manufacture, being much easier to make and providing a better bearing surface than a number of collars meshing with a spur gear.

Final Drive

The ten-tooth spiral bevel pinion on the gearbox driven shaft drives a crown wheel 195 mm. in diameter, attached by setscrews to a backing flange on the one-piece malleable-iron differential box. This is carried on taper roller bearings, one of which is mounted direct in the aluminium-alloy differential and gearcase housing. The opposite bearing, which being behind the crown wheel, is subject to more load, is carried in a steel sleeve in the aluminium-alloy cover. The short differential shafts, steadied by ball bearings, have large flanges splined to their outer ends, to which the cast-iron brake drums are bolted. Also secured to these flanges are the aluminium-alloy housings of the pot-type inner universal shafts. These are cylindrical and embrace cast-iron lantern pieces on which are machined the flat faces on which the die-blocks slide. The in-turned ends of the housings clamp the flanges of the rubber covers against the ends of the lanterns, which take their drive from lugs on the flanges fitting between the die-block guide faces.

The design of the universal is exceedingly ingenious, embodying as it does the absolute minimum of small parts. The only bolts required are those inevitably wanted to secure the pots to the flanges, while there is no awkward internal broaching of the guide faces and no loose wearing plates and securing screws. The case-hardened die-blocks work on needle roller bearings on hardened steel bushes pressed on to the arms of the crosspieces of the universal shafts. Attention should be drawn to the numerous drilled holes in the die-blocks, cross-pieces and bushes, presumably intended to promote circulation and supply of grease. It would seem, however, that the radial holes in the die-blocks will tend to allow the grease to be flung out by centrifugal force faster than it can work in through the lateral holes.

The short shafts integral with the universal-joint cross-pieces are splined into muffs that also fit the inner ends of the outer shafts. These pass right through the hubs and are secured by cone and nut to the jaws of needle-roller universals. The driven bearings of these are formed as blocks dowelled and studded to the outer faces of the hubs. The coupling muffs are slotted and clamped by bolts. On removal of these they slide laterally to separate the halves of the shafts and allow the differential assembly to be withdrawn without disturbing the road-wheels.

In order to permit the required angularity of the shafts the hub bearings are arranged to form unit opposed taper roller bearings. They are adjustable by ring-nuts and are housed in stampings forming the ends of the diagonally disposed "wishbones" of the rear independent suspension.

Rear Suspension

Two tubes, pressed and welded into the stampings carrying the hub bearings have, at their outer ends, flexible bushes forming the suspension bearings. These bushes are 75 mm. long with inner and outer rubber sleeve diameters of 20 and 40 mm. respectively. To limit the axial movement of the bushes which would result from the thickness of the rubber, they are both set at an angle with the line joining them. This line, the nominal axis of suspension swing, runs at an angle of 30 deg. with the transverse. The inner bushes are set at 20 deg. from this line and the outer ones at 10 deg. in the opposite direction. Thus forces set up by propulsive or braking loads which would otherwise appear as axial loads on the bushes are largely taken as radial loads. The considerable thickness of the rubber, which gives added cushioning between the wheels and the vehicle, is easily capable of taking the eccentric movements of the ends of the bushes due to their non-axial positioning.

The load is carried by coil springs seating on flanged conical pressings separated by rubber liners from the pressings that form the spring abutments. A bump rubber is attached to the pressings on the wishbones, which are secured by welding. The upper spring abutments are closed by a cover pressing which in turn seats on a thick rubber pad bonded to a thin pressing that seats in a pressed member of the chassis.

The extreme care taken to cushion the suspension by rubber as well as by steel springs is also to be observed in the linkage of the piston-type hydraulic shock absorbers. Not only are thick flexible bushes fitted to both ends of the link, but additional cushioning is provided in the link itself. This is in two parts, coupled by an axially-mounted rubber sleeve giving $\frac{1}{16}$ in. cushioned movement in both tension

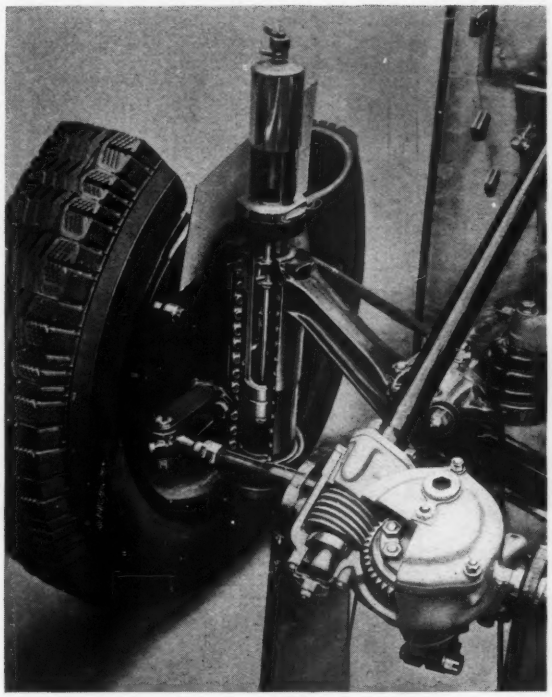
and compression before steel limiting washers take up the load.

A bump movement of 90 mm. is permitted and a rebound of 80 mm., these figures being considerably greater than those for the front suspension which are 70 mm. and 50 mm. respectively. The geometry of the suspension is a compromise between trailing-link and swing axle, on account of the diagonal location of the axis of movement. The roll centre is about 4in. above the ground while there is a "toe-in" effect from movements of the wishbones both above and below the normal giving some degree of "understeer" as the car rolls on a corner.

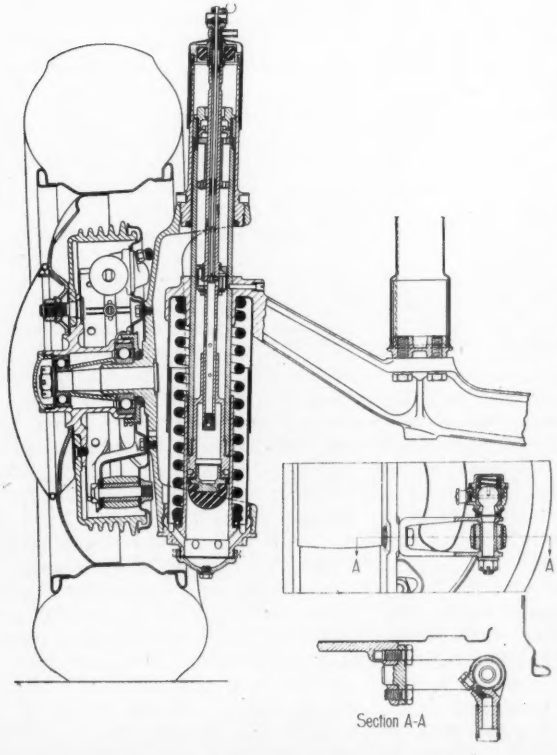
Front Suspension and Steering

The foundation of the front suspension is an "I" section beam stamping similar in appearance to an ordinary front axle with pads secured to the chassis structure by four setscrews each, picking up small pads welded to the chassis. Into an eye formed at each end of the beam is pressed a stout approximately vertical tube 40 mm. in external diameter. Each steering swivel comprises a light, wide jaw stamping carrying an upstanding tubular sleeve screwed to its upper eye and a somewhat similar sleeve projecting upward from the lower eye. These sleeves slide on the main tube, but the actual contact is limited to a short bushing at the top, actually separate from the tube itself and having a felt washer below it. A somewhat similar land, but without the felt washer forms the guiding surface at the lower end, both being made of bronze.

Thus the part of the tube that is exposed as the assembly slides, does not form part of the guiding surfaces, which are limited to the bronze members working on the protected inside surfaces of the upper and lower sleeves. The load is carried by a coil spring, seating on a light ball thrust bearing and enclosed in telescopic covers. Air displaced by the relative movement of the parts can escape into the loosely-fitting covers by diagonal holes. A loosely-fitting helmet pressing gives similar facilities at the upper end.



Sectioned front suspension and steering mechanism.



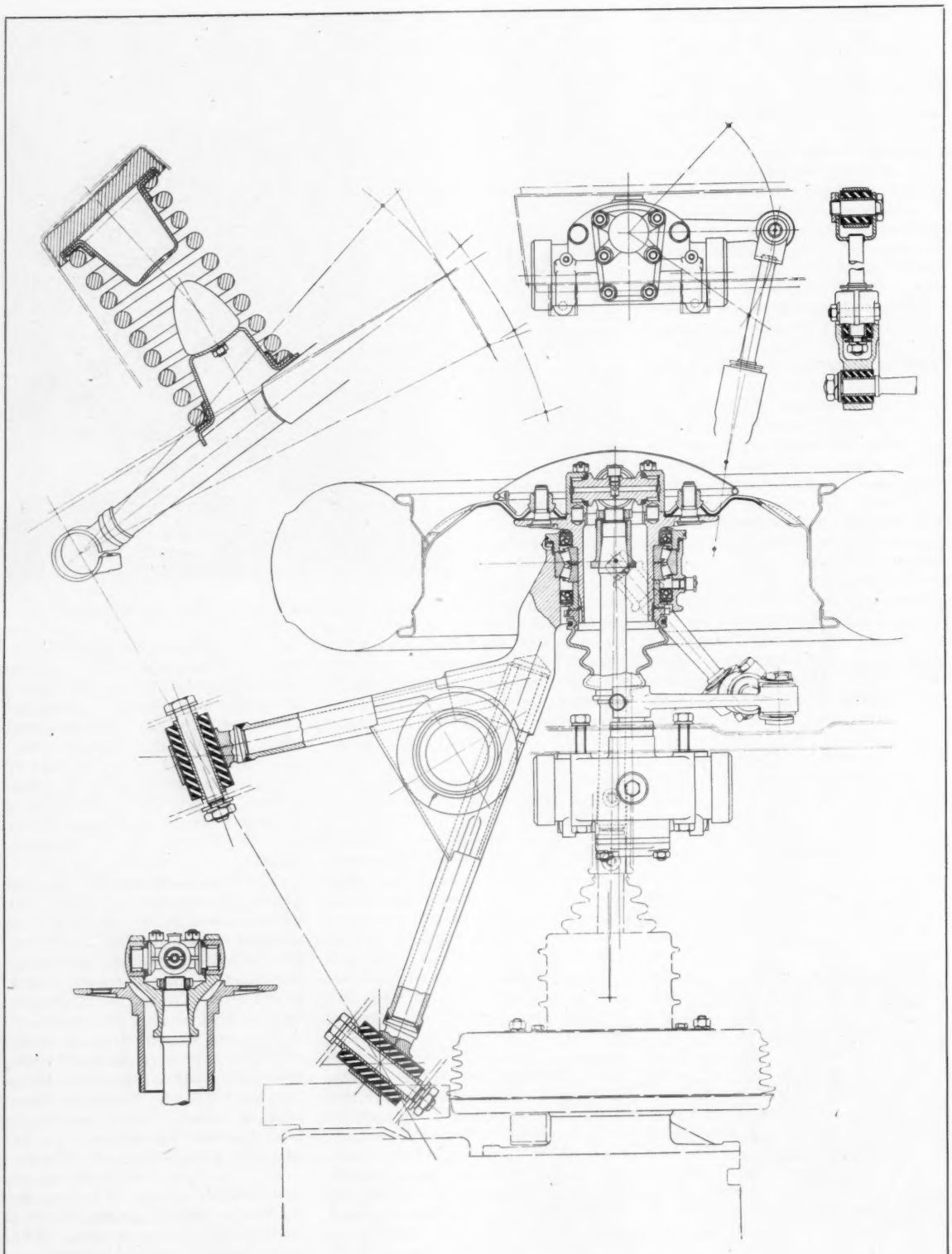
Vertical pillar independent front suspension.

The inside of the main tube forms the working cylinder of a double-acting hydraulic shock absorber, the piston rod of which, is attached to a plug closing the end of the upper sleeve screwed to the swivel. Control is by flat spring valves giving two-way damping with an orifice by-passing the valves in both directions of flow. It can be regulated by a screwed needle-valve.

Since there is no external containing tube as in the orthodox telescopic shock absorber the recuperation chamber is perforce an extension of the working cylinder. The displacement of fluid by the piston rod sets another problem. This is dealt with by providing an extension of the rod below the piston which runs through a bush into a chamber. The hollow piston rod carries a ball valve at its lower end, which, on the downward stroke passes oil up the rod, while on the upward stroke a vacuum is created. In a short distance of running a condition is established where a vacuum is built up and displacement of oil practically ceases.

A reservoir is provided on the scuttle of the car containing thin engine oil and piped to the top of the absorber assembly. Oil can be supplied from this to make up for leakage, any excess running down the outside of the main tube and lubricating the guide. The lower guide, being inverted, can be relied on to retain enough oil for lubrication over considerable periods. The main tube is not quite vertical, being set at an angle of 1 degree 7 minutes and at a negative castor angle of 1 degree. The wheels have about 2 degrees of camber.

Since the upper end of the suspension assembly necessarily projects past the tyre the offset at ground contact is considerable, being 84 mm. While this is probably of little importance in ordinary driving, considerable reaction at the steering seems likely when braking hard on a surface so irregular as momentarily to unload one tyre or the other to the point of skidding and reduction of brake drag. From the point of view of steering geometry the



LAYOUT OF INDEPENDENT REAR SUSPENSION.

system gives a roll centre practically at ground level, while the wheels roll with the car, giving a tendency to understeer.

Theoretically, there are changes in "toe-in" under various suspension movements. The swivels are joined by a track rod 1,162 mm. long, while the steering arms have a radius of 150 mm. Displacement of one swivel by 50 mm. would only cause a foreshortening effect on the track rod of about 1 mm., giving a change of "toe-in" of about one-sixth of a degree per side, a negligible figure.

The employment of rubber bushes of considerable radial thickness (about 5 mm.) for the track-rod joints is of interest. On account of the 84 mm. ground-contact offset there will be considerable tension in the track-rod when braking and each bush might perhaps move eccentric by say 1.5 mm. or 3 mm. for the two. But this again is only about 1 deg. toe-out measured at one wheel, which again is not of serious importance. In any case the actual handling of the car at all speeds is stated to be irreproachable, and there is no evidence that tyre wear is excessive.

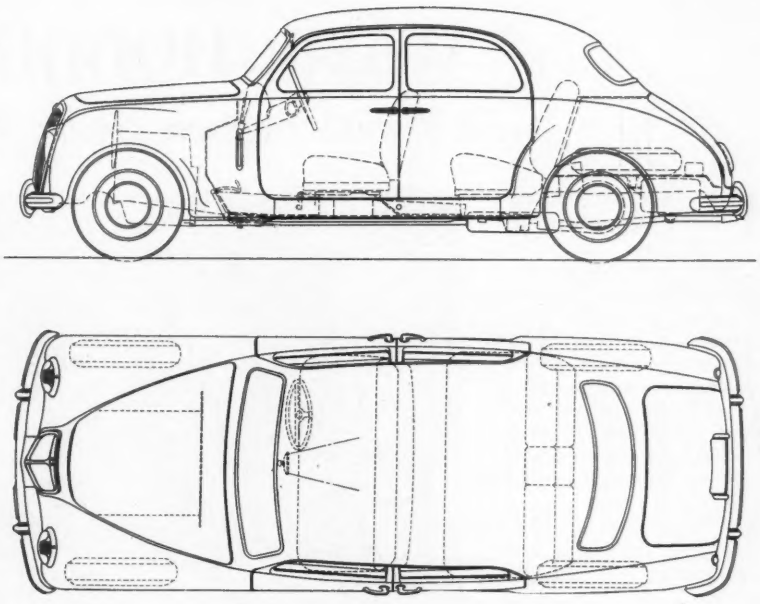
The steering box is mounted in front of the suspension beam and is connected to the opposite swivel by a long rod with orthodox ball joints. The drop-arm spindle is mounted vertically, eliminating "rise and fall" errors on the drag-link. The inclined column spindle carries a worm that has such a thread angle that the appropriate wheel on the drop arm spindle becomes a straight spur gear. The contact between the steel worm and the bronze spur gear is not as good as for a worm and worm wheel but the parts are amply dimensioned. Should any wear occur on the gear teeth, the wheel can be turned to bring new teeth into use.

Brakes

In contrast with English practice, 2LS brakes are not employed. The cast-iron brake drums are 275 mm. in diameter with a facing width of 60 mm. and have five ribs, with a wide angle of taper, increasing in depth from the outer to the inner edge. Die-cast aluminium alloy shoes are employed, mounted on individual anchor pins of casehardened steel working direct in the shoe metal. Those for the front brakes take the form of sleeves, nipped between the brake back plate pressing and a second stout pressing. All the back-plate setscrews are used for its attachment and it is inswept at its centre to serve as a grease catcher, the securing means being a throughgoing 10 mm. bolt.

Floating cam action is used, and the hydraulic cylinders for the front brakes are larger than those at the rear, giving a braking ratio of about 60 to 40, front and rear. Clearance adjustment is by snail cams and the pierced disc wheels give good ventilation of the front drums. The rear drums are mounted on the differential assembly and are in the air stream passing under the car so that they should be adequately cooled.

Hand brake operation is by hanging levers integral with short cross-shafts, the outer ends of which are spherical and work in holes in the webs of the front shoes near the tips. The inner ends of the shafts are recessed and turn on spherical pins secured to the differential housing, the bearing point being almost exactly in line with the hanging levers. The pull of the brake cable has therefore no direct reaction on the front shoes. Expansion of the shoes is by short upstanding arms integral and counterbored to take the spherical ends of rods reaching to the rear shoes. Application of the handbrake is by cables connected to a lever hanging down from the top of the scuttle assembly.



Lancia "Aurelia" standard saloon.

General

The standard car has integral body and chassis construction with box-section side rails and box-section roof rails also. The contribution to general stiffness made by the very strong roof is somewhat reduced by the fact that a pillarless construction is used for the doors, which latch on to floating tapered pins at top and bottom. The floor is integral and is level at a height of about 10in. from the road, being well below the body sills.

On account of the rear location of the clutch and gearbox, the floor is but slightly obstructed by tunnels and the front footboards are clear of the clutch housing and gearbox. The scuttle and the front seat supports form an integral part of the cross-bracing and the structure is stiff enough to dispense with the assistance of the body itself. The vehicle is in fact also offered in chassis form, *i.e.*, with body sills, scuttle and floor for the fitting of individual bodywork by specialist coachbuilders.

Dimensions

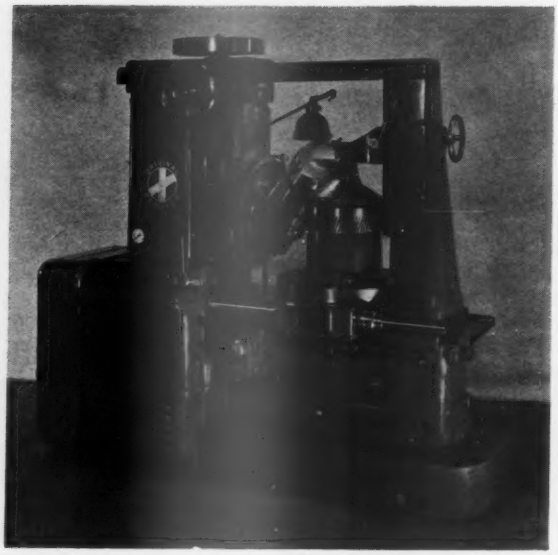
In standard form the vehicle has a dry weight of 21¼ cwt., a remarkably low figure for a car giving reasonable accommodation for six passengers. The overall height is only 4ft. 11ins. the overall width 5ft. 1in., the wheelbase 9ft. 4½ins., the front track 4ft. 2½ins. and the rear track 4ft. 3ins. A turning circle of 32ft. is provided, with about four turns from lock to lock.

The minimum ground clearance is 6ins., but this figure is under the gearbox and rear brake drums, which are sprung parts. At the front the clearance is 6¾ins. The fuel capacity is 13¼ gallons, including 1¼ gallons reserve, the cooling system holds 18½ pints and the sump 7 pints. The standard tyres are 165 × 400 on wide base rims; for export 5.50 × 16 tyres are offered.

GEAR HOBBING

A Recently Developed Pfauter Machine

IN pre-war days the motor industry made widespread use of hobbing machines designed and manufactured by Herman Pfauter Machine Tool Works, Konrwestheim, Germany, and indeed many of these pre-war machines are still operating satisfactorily in British factories. Post-war design Pfauter gear hobbing machines are now available in a wide range of sizes and types. These notes are concerned only with the re-designed RS1 vertical hobbing machine, which is illustrated in the accompanying photograph. It will cut spur gears, worm wheels, chain wheels, splines, serrations and all other forms that can be hobbed on cylindrical workpieces up to 3 D.P. or equivalent cross section of tooth space.



Pfauter RS1 Gear Hobber.

This machine is designed to give high production rates with great accuracy and a high standard of finish. For this reason special attention has been paid to rigidity. Not only is the upright extremely substantial, but the method of mounting the workpiece by means of a work arbor and an overarm type work arbor support ensures accuracy even under the heaviest cuts.

The work table is indexed through a worm gear drive, an arrangement that is standard on all machines in the Pfauter range. Any backlash that may develop between the worm and the wormwheel can be readily compensated for without alteration of the distance between the axes of the worm and wormwheel. The worm is constructed with a progressive thickening of the tooth form so that play can be taken up by axial adjustment of the worm. Other improvements included in the design are a trip dog attachment with micrometer adjustment for controlling the depth of cut, and radial feed equipment with a drop worm for hobbing against a positive stop. The hob spindle bearings are also radially and axially adjustable to obtain accurate tooth form.

This machine may be used for either conventional hobbing or for climb hobbing. Where possible it is advantageous to use the climb hobbing method. This gives shorter production times since it allows a faster cutting speed and an increased feed to be employed. It also gives a better finish on the work because it produces a better chip form and a longer tool life. The hob saddle feed is operated by means of an hydraulic control unit. A single high-pressure pump supplies oil to this control unit and also circulates oil throughout the lubrication system. Feed and quick motion of the hob saddle or workpiece table are operated by a single lever control.

The main drive is taken from a 4 h.p., 1500 r.p.m. three-phase squirrel-cage induction motor, and the quick traverse is driven by an independent 1 h.p., 1500 r.p.m. motor of the same type. Both units are fitted with a starting torque damper. Provision is made for hob speeds from 45 to 180 r.p.m. in six steps. Workpieces up to 29½in. diameter and 12⅜in. width can be accommodated. The maximum diameter and length of hob that can be accommodated are 4⅛in. and 5½in. respectively, and axial movement up to 2in. is permissible. The distance between the hob and the work arbor can be adjusted between 1 3/16in. and 17⅜in.

Spur gear hobbing is effected by feeding parallel to the axis of the work, with the hob and the gear rotating in relation to each other like two meshing gears. For hobbing helical gears additional equipment is necessary to give a supplementary rotation to the work corresponding to the tooth angle. This operates by means of a differential drive with a corresponding change gear system. Mating helical gears can be hobbed, independent of the number of teeth and the feed, without any alteration of the differential change gears. Therefore, such gears are produced with parallel axes and uniform engagement is ensured.

Worm wheels with tooth angles up to 6 or 8 deg. are hobbed with the work fed radially to the hob. For hobbing worm wheels of higher tooth angle the tool is fed tangentially to the work by use of the crossfeed attachment and differential drive used for hobbing helical gears. This method is also used for producing worm wheels with a fly cutter and for producing worm spindles with a pinion type cutter. The sole agents in this country for this machine and other Pfauter hobbing machines are Vaughan Associates Limited, 4 Curzon St., London, W.1.

Nimonic 90

AN addition to the Nimonic series of alloys has recently been announced by the Mond Nickel Co. Ltd., Sunderland House, Curzon Street, London, W.1, and it has been in regular production for some time by Henry Wiggin & Co. Ltd. Known as Nimonic 90, this new alloy has already been used for a considerable time in several standard gas turbine engines. It is a wrought alloy that may be operated at higher temperatures and stresses than the previous best wrought material available, namely Nimonic 80A. Creep criteria specified in the appropriate Ministry of Supply specifications show that at 750 deg. C. Nimonic 90 is required to withstand a stress of 19 tons per sq. in. compared with the 17 tons per sq. in. required of Nimonic 80A. It is stated that at 815 deg. C. and even 870 deg. C., Nimonic 90 has shown high load carrying capacity for long periods. It is available in bar sections, forgings, sheet, strip and wire. (1941)

BRAKE DESIGN

A Detailed Investigation of Fade Sensitivity

By G. A. G. Fazekas.

After deriving a yardstick of "μ consciousness" that is complementary to shoe factor, it is shown that, other things being equal, the two are in straight proportion, and that the frictional qualities of linings should be judged by an "index" that takes into account both fade and the absolute value of μ. This revealed that the inherent "μ consciousness" of fixed cam, duo-servo, two-leading and leading-trailing shoe brakes is in the order shown.

APART from the hook-up, problems of brake design may be perhaps divided into two groups, (i) shoe geometry, and (ii) brake geometry, *i.e.*, the combination of shoes into various brakes. With regard to both, the usual shoe factor analysis supplies reliable design data so far as torque output is concerned, on which, however, there is an upper limit imposed by inherent "μ consciousness". Although this is well realised, beyond some occasional allusions to the slope of the shoe factor curve, which may be very deceptive, there does not seem to be any direct yardstick for "μ consciousness" that is comparable and complementary to the shoe factor. It is to fill this need that the present paper has been compiled, the treatment being confined to what is thought to be a novel interpretation of the shoe factor graph with some of its implications and applications.

These were fully confirmed in the author's experiences, although it has not yet been possible to compile experimental evidence fit for scientific consumption. All that can be said is, that the accuracy of the design data given, is about the same as that of the shoe factor analysis. The subject matter itself comprises two parts. The first deals with the derivation of a scale for, and the implications of, fade sensitivity. The second shows the application to some brake geometries with a view to assessing their inherent merits from the point of view of geometry alone. This, of course, is only a part, but not an unimportant one, of the whole picture.

The following abbreviations are employed in the text.

$$\text{Shoe factor} = \frac{\text{drum drag/shoe}}{\text{shoe tip load}}$$

$$\text{Torque factor} = \frac{\text{sum of shoe factors/brake}}{\text{total drum drag}}$$

$$\text{Brake factor} = \frac{\text{total drum drag}}{\text{total nominal input load}}$$

$$\text{Fade} = \frac{\text{change of torque}}{\text{nominal value of torque}}$$

$$\text{Sensitivity} = \frac{\%\ \text{change of torque}}{1\%\ \text{change in friction}}$$

Brake factor includes thus all expander friction, return springs, etc., whereas shoe and torque factors do not.

The various brake geometries will be designated as :

2L = two-leading shoe brake. See Fig. 13.
LT = leading—trailing shoe brake. See Fig. 6.
DS = duo servo brake. See Fig. 10.
NS = fixed cam brake. See Fig. 15.
SS = self servo brake. See Fig. 14.
3L = three-leading shoe. See Fig. 9.

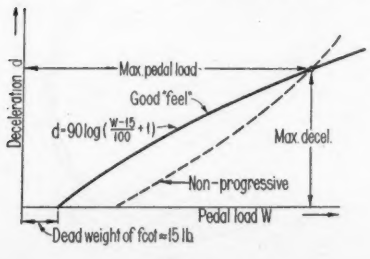


Fig. 1

Physical basis of sensitivity

When the friction of a brake lining changes from its nominal value (μ) to say (μ — Δμ), and if the pedal load (W) remains constant, the brake torque will fade from its nominal value (T) to (T — ΔT). It is now submitted that (ΔT/T) seems a true yardstick of fade for the two following reasons :

(i) Physically, (ΔT/T) is a direct measure of the relative change in torque, deceleration, stopping distance, stopping time, regardless of weight, speed and absolute deceleration of the vehicle. Like the shoe factor it is dimensionless and not tied to any specific application.

(ii) Physiologically, (ΔT/T) is also an approximate measure of the sensation of fade, as experienced by a driver. This can be explained by the Weber-Fechner law which states that human sensations (σ) are in straight proportion to the logarithm of the stimuli, which in this case is pedal load (W) or deceleration (d); *i.e.*

$$\sigma' \propto \text{log. W or } \sigma'' \propto \text{log. d} \quad (1)$$

It follows therefore, that a small change in (σ) is to the first order of small quantities.

$$\Delta\sigma' = \frac{\Delta W}{W} \approx \frac{\Delta T}{T} \text{ for d = constant} \quad (2)$$

$$\Delta\sigma'' = \frac{\Delta d}{d} = \frac{\Delta T}{T} \text{ for T = constant} \quad (3)$$

The author obtained some indirect support for this reasoning, viz., the applicability of the Weber-Fechner law, from the repeated observation that if the pedal load vs. deceleration characteristic of a vehicle is as shown in heavy line on Fig. 1 the "feel" of the brake is good. Conversely, the opposite, as indicated by a broken line, results in a "non-progressive" brake, even though the maximum performance figures be the same. Incidentally, both characteristics are primarily due to linkage layouts and/or too heavy return springs.

If (ΔT/T) is thus adopted as a yardstick of fade, it has to be considered that (ΔT) itself is due to two factors, (i) the actual change in friction (Δμ) which is (ii) amplified through the self-servo effect of a shoe. Since it is attempted here to assess only the intrinsic reactions of the brake, clearly (Δμ) has to be supposed the same in all cases. (It is an entirely different matter that some brakes generate more or less (Δμ) under otherwise identical duty). Further, and to cater for different brake linings, it appeared more appropriate to use not (Δμ), but its relative value (f = Δμ/μ) as the common denominator.

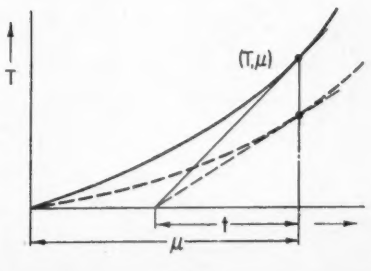


Fig. 2.

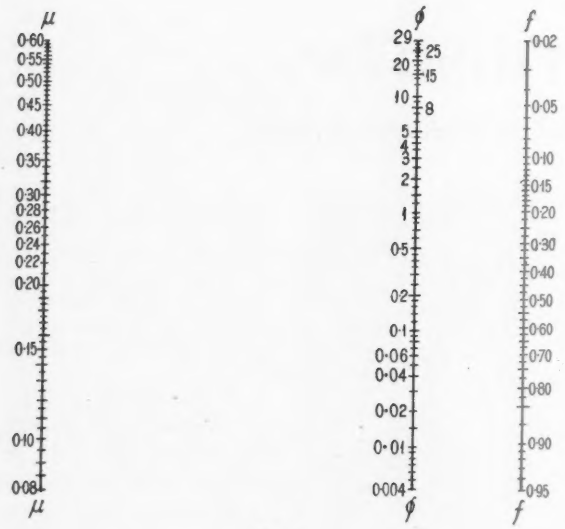


Fig. 3.

The fade or the "grab" of a brake caused by any change of friction, (f), may be thus assigned the numerical value of

$$\overline{S} = | \Delta T/T | \qquad (4)$$

If now Fig. 2 represents a typical graph of torque vs. friction for a brake, on supposing that (f), and hence (ΔT) are both small, the tangent can be substituted locally for the curve and the sensitivity (S) defined as

$$S = \lim_{f \to o} (\overline{S}/f) = \mu/t \qquad (5)$$

(S) states thus how much faster the torque changes than (μ), (its practical values are about equal to shoe factor). Although equ. 5 is strictly valid for small (f) only, nevertheless, it gives a good indication for larger values, too, and in the author's experience, proved itself very well for comparative assessment.

That the slope alone is rather misleading is also shown on Fig. 2 where both T vs. μ curves have the same sensitivity, although the slopes differ very much. That this must be so will be obvious if it is remembered that otherwise the μ-consciousness of a brake could be altered at will merely by changing the pedal ratio.

The sensitivity thus defined seems to be closely allied to what is sometimes referred to (Vallance and Doughtie, 1943) as servo gain, and defined as the increase of pressure between shoe and drum, generated by servo action. It is shown in Appendix 1c that if the force between shoe and drum is (Fo) for μ = o, and (F) for (μ = μ), then with fixed anchor shoes :—

$$\text{Servo gain} = F/Fo = S.\sqrt{1+\mu^2} \approx S. \qquad (6)$$

This applies however, to fixed anchor shoes only, whose centre of drum drag does not vary with (μ), so that (shoe tip travel)/(shoe centre lift) is independent of (μ). On the other hand floating shoes — that have two degrees of freedom of movement — vary their wear patterns (Acres, 1946), with (μ) which, of course, alters (shoe tip travel)/(shoe centre lift). Therefore (F) and (Fo) refer to different leverages, as it were, the physical significance of (F/Fo) becoming thus somewhat nebulous. Clearly, no such limitation applies to sensitivity as defined before, besides which it is a direct measure, and as will be seen, it is easily computed, too.

Application to Brake Assemblies

For a complete brake installation, with known details, a resultant torque vs. μ graph can be easily, though laboriously compounded for a certain pedal load (its choice is shown to be immaterial) and the overall sensitivity obtained by the construction shown on Fig. 2. Fortunately, this process can be avoided, and it is shown in Appendices Ic and II that only the following steps are necessary.

(a) The shoe factor curve of any brake can be readily obtained (*e.g.*, by graphical analysis), and the process further simplified by using a special graph paper to inverse scale, as on Fig. 7, which, for reasons explained in Appendix Ic, results in straight line graphs.

(b) It is further shown in Appendix Ic that the (S) vs. (μ) graph is also a straight line to a semi-inverse scale, as shown on Fig. 8. Furthermore, for all shoes S = 1 at μ = 0, so that the graphical construction indicated on Fig. 2 need be carried out for one point only. Actually, this second point can be more easily obtained by measuring the slope (r) of the shoe factor line on the inverse scale graph, and then the sensitivity (S) is given for a particular value of (μ) and (K) as

$$S = r.K/\mu \qquad (7)$$

This construction is generally valid for all inverse scale shoe or torque factor graphs, whatever their shape.

(c) From the above graphs, the sensitivities (S) and shoe factors (K) being known for each shoe, it is explained in Appendix II that the overall sensitivity (So) of the whole brake can be calculated by the formula

$$So = \frac{S' \,.\, K' + S'' \,.\, K''}{K' + K''} \qquad (8)$$

the accents referring to the individual shoes, the mechanical advantage of the hook-up being irrelevant. Further, if both shoes are identical

$$So = S' = S'' \qquad (9)$$

Alternatively, the torque factor curve may be plotted—in general it is not a straight line—and the overall sensitivity obtained by the method shown in (b) ante.

(d) When the sensitivity of the entire braking system (Soo) is required, for example, with different front and rear brakes, actuated through lever ratios (m′), (m″) with (Ko′) and (Ko″) as the torque factors, it is also shown in Appendix II that

$$Soo = \frac{m'.So'Ko' + m''.So''.Ko''}{m' \,.Ko' + m'' \,.Ko''} \qquad (10)$$

the accents indicating here different assemblies.

The straight line graphs mentioned are strictly true for single fixed anchor shoes only, both the torque factor and the overall sensitivity resulting in slightly "bent" straight lines if the shoes are not the same. As, however, it is far easier to work with "bent straight lines" than with curves proper, the inverse scale graph can be recommended for general use.

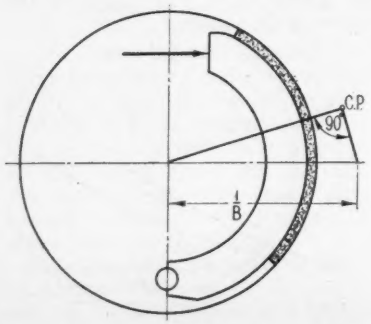


Fig. 4.

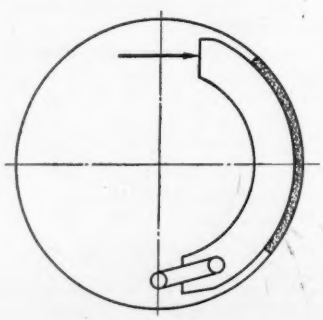


Fig. 5.

The shoe factor curves of non-fixed anchor brakes being in general cubic hyperbolae, it can be strictly proven that the inverse scale graphs are not straight lines. Fortunately, however, this cubistic influence does not seem to have penetrated very far so that, in practice, no measurable deviation can be found.

Some basic properties of brakes

There are certain fundamental limitations to what a brake shoe can do, and for fixed anchor shoes it is shown in appendix Ic that all the basic properties can be summed up in a single equation :

$$K = \mu.(1/f - 1).\frac{s}{1-s}.\frac{A}{B} = \phi.\frac{s}{1-s}.\frac{A}{B} \quad (11)$$

where K = nominal shoe factor, corresponding to
μ = nominal lining friction.
$\bar{s}$ = actual relative change of torque ($\Delta T/T$) caused by
$f = \Delta\mu/\mu$ = relative fade of lining friction.
$\phi = \mu.(1/f - 1)$ = lining index.
A = (shoe tip travel) / (shoe centre lift).
B = a factor depending on length of lining arc and wear pattern only ; its meaning is illustrated on Fig. 4.

This seems the main functional equation of a single shoe, be it leading or trailing, and both for fade and "grab". Care has to be taken, however, in interpreting the signs of ($\bar{s}$) and (f) : in the form shown equ. 11 is valid only for the fade of a leading shoe. For "grab" the signs of both (f) and ($\bar{s}$), and for a trailing shoe only that of (f), have to be reversed.

The practical implications of equ. 11 appear thus :

(i) two linings are equivalent under the same duty if they have the same lining index (ϕ) as defined by

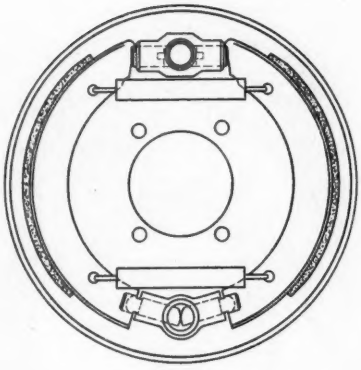


Fig. 6.

$$\phi = \mu.(1/f - 1) \quad (12)$$

which is illustrated by the nomogram Fig. 3. For example, a lining characterised by $\mu = .40$, $f = 12\%$ and $\phi = 2.95$ yields a brake which is no better and no worse than one adapted to a nominal lining friction of $\mu = .123$, but with 4% fade only. (Such "adaptation" of the design would be carried out by changing the anchor position, and the position of shoe tip actuation to keep (A) constant). As mentioned, equ. 12 is valid for fade only ; and for grab, *i.e.*, an increase in friction, it should read

$$\phi' = (1 + 1/f).\mu \quad (13)$$

For trailing shoes the opposite applies. In the present practice, of course, only leading shoe fade seems to be of importance.

(ii) If (f) is small, Equ. 11 can be written as

$$K = \mu.S.A/B. \quad (14)$$

where (S) is the sensitivity factor, so that approximately

$$K \propto S.\mu. \quad (15)$$

When other things are equal (μ) thus assumes the significance of (ϕ). This is equally valid for leading or trailing shoes, and for fade or grab.

(iii) Long unsymmetrical lining arc is preferable by geometry : in practice, of course, this may be more than offset by the consequences of too concentrated wear.

As mentioned, the foregoing is strictly valid for fixed anchor shoes only, as indicated diagrammatically on Fig. 4, and must be regarded only as a reasonable approximation for floating shoes, *i.e.*, shoes that have two degrees of freedom of movement, as indicated on Fig. 5.

Another reservation is that the toe-grab angle was disregarded in the calculations that led to these results. With low and medium duty brakes this is quite justified, but the author found evidence that toe grab has a profound influence on the judder of hot running brakes.

Basis of comparison between various brakes

Ultimately the performance of any brake installation is judged in terms of pedal load, pedal travel, rate of adjustment for lining wear, and fade. Of these four quantities three have to be the same, actually the fourth serving as a criterion. This may be, perhaps, reasonably approximated by imagining a vehicle to be fitted with a certain brake that is then replaced by a 2L type designed to have the same

torque factor .. .. .. K.
(shoe tip travel)/(shoe centre lift) A.
lining material (μ) .. } ϕ
friction fade ($f = \Delta\mu/\mu$) }
lining arc and wear pattern .. B

as the brake it is compared with. On this premise, the question of superiority is reduced to finding the corresponding sensitivity factors according to equ. 14. ; but temporarily the reservation has to be made that neither fade nor total wear will be affected by the work distribution between the shoes. The investigation has been further simplified by restricting it to fixed anchor shoe brakes, although not all brakes lend themselves readily to mounting this type of shoe. Still, the difference between fixed and

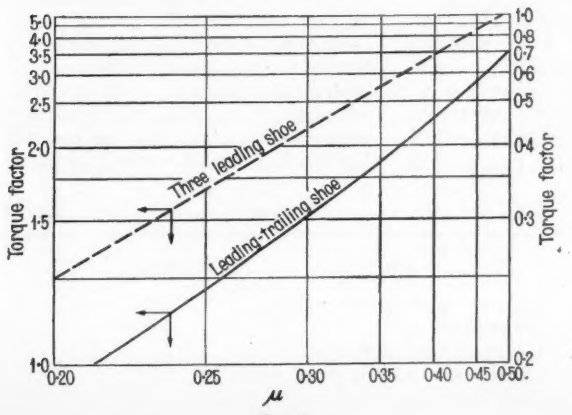


Fig. 7.

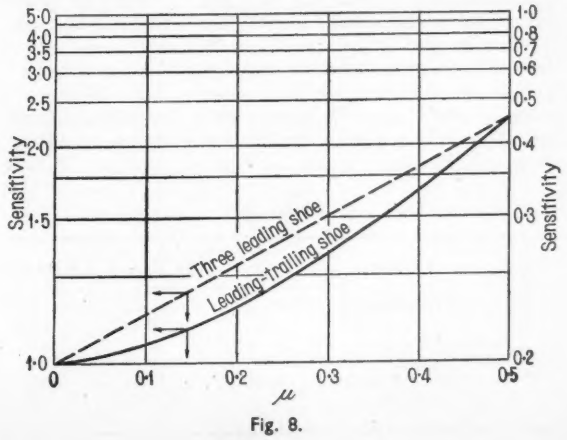


Fig. 8.

floating shoe brakes being in the order of some 14% on the sensitivity basis in favour of floating shoes, this is not big enough to influence findings materially. On this basis the 2L. brake is supposed to be levelled by suitably positioning its anchor pins and wheel cylinders, toe and/or heel grab being discounted.

3L. vs. LT. Brake

A modern LT. brake mounting two identical fixed anchor shoes is shown on Fig. 6 with the corresponding torque and sensitivity factors in heavy line on Figs. 7 and 8 respectively. By way of comparison with actual practice, the illustration Fig. 9 shows a 3L. racing car brake that the author has designed for the data shown in broken lines on Figs. 7 and 8, a summary of the salient points being given in Table I.

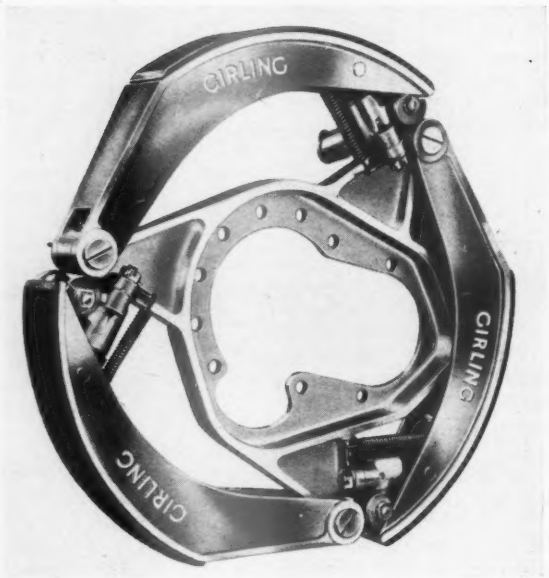


Fig. 9.

The stipulated common denominator is but roughly achieved in this instance, the scales being tipped in favour of the LT brake, so that it is necessary to bring matters first to a common denominator. If now the 3L brake's (shoe tip travel)/(shoe centre lift) is decreased to 3.90, its shoe factor would be $2.65 \times 3.90/4.50 = 2.30$, but its sensitivity would remain unaltered at 1.75. On the other hand, if the 3L brake would mount the same lining as the LT, as stipulated, its sensitivity should be in accordance with equ. 14, $1.75 \times .34/.40 = 1.49$, which is some 10% lower than for the LT brake. (In this connection it has to be remembered that as stated earlier, (μ) is really synonymous with (ϕ)).

This is no coincidence, and it is strictly proven in Appendix IV (a) that on the stated premises, an LT brake is perforce always inferior to a 2L or 3L type, quite apart from the fact that owing to equal work distribution between the shoes, with the 2L or 3L there is bound to be less fade and wear. This seems, at first, contrary to usual practice, but it may be remembered that if an LT and a 2L brake, both mounting identical shoes, are compared, the 2L brake is actually much more powerful, probably by some 50%. If now the power of the LT brake were raised to the level of a 2L, clearly most of the increase could come only from the leading shoe, which would then become comparatively unstable on account of its high power.

DS. vs. 2L. Brake

The construction of these brakes is usually as shown on Fig. 10, *i.e.*, both shoes are identical, but in operation the primary is a floating and the secondary a fixed anchor shoe. The corresponding torque factor and sensitivity graphs are in heavy line on Figs. 11 and 12, together with those of a modern 2L brake, shown in broken lines, the brake itself being illustrated on Fig. 13. A summary of the main data is given in Table II.

This would appear to show some 12½% more fade sensitivity for the DS. brake. Actually, however, the picture is deceptive because the 2L brake in question is again fitted with high, and the DS. brake with low friction linings. When allowance is made for this, as before, the corresponding sensitivity figure of the 2L brake would be $2.0 \times 0.40 \times 3.95/(0.305 \times 4.43) = 2.34$, *i.e.*, the DS. brake would appear to be intrinsically superior. That this is no coincidence is shown in Appendix IV (b) where a general proof is given for a somewhat simplified arrangement.

It is an altogether different problem that, in practice, the DS. brake is not too suitable for high friction linings, whereas the 2L type is, having the added advantage of equal work distribution between the shoes. On the other hand, the lining index of low friction linings is often superior, compared with high friction linings, and this, coupled with efficient hook-up, may explain the popularity that DS. brakes enjoy in the U.S. Conversely, the root cause of frequent trouble can be an altogether too high torque factor that would cause instability on any other brake.

It would thus seem that the shoe factor characteristic of a DS. brake is not so bad as is sometimes suspected. The reason lies in the fact that when taken singly, and if properly designed, both shoes of the DS. brake have lower shoe factors than those of the comparable 2L brake. The power of the DS. brake originates therefore, from the combination of a large, but only partly "μ conscious" force actuating a very stable shoe. At the same time it has to be realised that the DS. brake is fundamentally best suited for high power, and it is shown in Appendix IV (b) that this superiority increases with torque factor.

SS. vs. 2L. Brake

Every now and then attempts are made to utilise part of the anchor reaction of a brake as an additional shoe application force, resulting in a kind of self-servo effect. Fundamentally, this type of brake is as illustrated on Fig. 14, where the anchor pin is carried by a heavy lever, which is pivoted on the back-plate, its other end pushing the shoe tip. It is shown in Appendix IV (c), that this is neither better nor worse than the conventional fixed anchor type, in spite of the added

Table I

Data	3L Brake	LT Brake
Shoe tip travel/shoe centre lift ..	3 × 1.50 = 4.50	2 × 1.950 = 3.90
Designed for μ	.34	.40
Corresponding torque factor ..	2.65	2.30
Corresponding sensitivity factor ..	1.75	1.65
Total lining arc	3 × 73°	2 × 100°

Table II

Data	DS.	2L.
Shoe tip travel/shoe centre lift ..	4.43	2 × 1.975
Designed for μ = ...	.305	.40
Corresponding torque factor ..	3.5	3.5
Corresponding overall sensitivity ..	2.25	2.00
Total lining arc	2 × 115°	2 × 100°

complications, and discounting the inevitable elastic deflection of the lever, or its equivalent. This would probably explain its conspicuous absence in practice. The explanation seems to be that the additional lever increases both power and lost motions, the two exactly cancelling each other, as shown in Appendix III (b).

NS. vs. 2L. Brake

A diagrammatic arrangement of this type is shown on Fig. 15, the expanding system being such that the shoe tips are constrained to equal movements. If then the displacement vs. load characteristic obeys Hooke's law, it is shown in Appendix III that the torque factor is

$$Ko = 4.K' \,.\, K''/(K' + K'') \quad (16)$$

where (K′) and (K″) denote the shoe factors of the leading and trailing shoes if taken singly. As is known, this results in a linear relationship between (Ko) and (μ), when the overall sensitivity = 1 for any value of (μ). It would thus follow that since the sensitivity of a 2L brake is always greater than unity, the NS. brake is superior.

Actually, however, both are equal if the stipulated basis of comparison is adhered to, for it is shown in Appendix IV (d), that the comparable 2L brake degenerates into one without any servo action. In practice this can be realised by putting the (virtual) anchor on the line of drum drag ; with symmetrically disposed linings this results in parallel shoe movement. On the other hand (shoe tip travel)/(shoe centre lift) = 1 for such a brake, whereas it is about 2 for the NS. type, so that the two can be brought to equal torque factor basis only by embodying a 2 : 1 lever, or like in the equivalent, but degenerate 2L brake. It is perhaps more appropriate to state that the two brakes seem incommensurate, except by mathematical trickery.

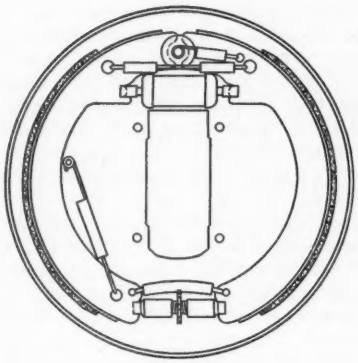


Fig. 10.

Incidentally it may be also mentioned that brakes with parallel shoe movement are sometimes used for friction tests owing to the "theoretical" absence of servo action. But theory suggests that parallel motion alone does not necessarily exclude servo action. (In fact the brake may be brought to self-locking if the lining geometry is wrong, and/or the lining is very inhomogeneous.)

Whereas the brake does not appear thus too suitable for friction tests, perhaps it could be developed as an alternative to the conventional NS. type, having the advantage that highly efficient floating cams (hydraulic) could be used.

Summary and conclusions

In Lord Kelvin's view a phenomenon cannot be understood until it is measured appropriately. It was in an attempt to comply with this philosophy that the author proposed as a tentative measure of

$$\text{fade} = \frac{\text{actual deviation of torque}}{\text{nominal value}}$$

some justification for which was found both on physical and physiological grounds.

In dealing first with the physical aspects, it has been strictly proven, but for fixed anchor shoes only, that the output one can obtain from a shoe depends on four factors: (i) a geometrical one uniting the effects of lining arc and wear pattern, (ii) (shoe tip travel)/(shoe centre lift), (iii) fade as defined above and (iv) a lining index that sums up in one measure the frictional properties of the lining. The existence of this fundamental relationship shows then clearly the limitations, and suggests that the value of a lining depends not only on high (μ) but also on its stability. This will be also largely if not numerically true of floating shoes.

For practical design work the notion of the fade factor has been simplified and

$$\text{sensitivity} = \frac{\%\text{ change of torque}}{1\%\text{ change of }(\mu)}$$

adopted as a reasonable approximation, complementary to shoe factor for characterising a brake. For practical work it was further found that analysis is simplified, both shoe and sensitivity factors resulting in straight, or substantially straight line plots, when inverse and semi-inverse scales respectively are used as co-ordinates.

On comparing then the relative merits of various brakes, the problem was viewed as follows: Let it be supposed that a vehicle is fitted with a certain brake, resulting in a certain pedal load, pedal travel, fade and lining wear. On replacing now this brake with a 2L type of the same size, but so made as to be identical in everything but fade, the latter may serve as a criterion of superiority. The problem was then simplified by considering fixed anchor shoes only, stipulating the same lining material, and assuming that friction fade is not affected by such difference in work distribution between the shoes as may exist.

On this basis it transpired that, in

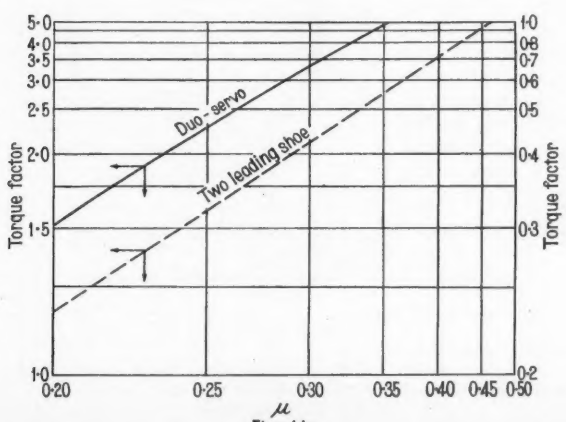


Fig. 11.

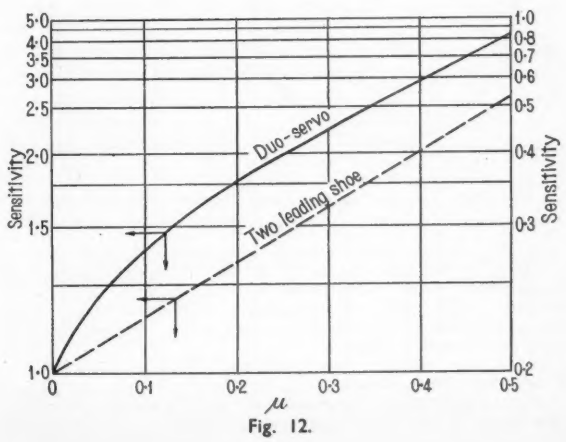


Fig. 12.

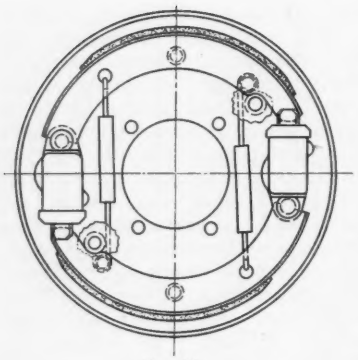


Fig. 13.

present practice, the most stable is the fixed cam type, the equivalent 2L brake degenerating into one with parallel moving shoes. As these may embody a highly efficient floating cam, mechanical or hydraulic, perhaps its practical scope could be extended beyond friction tests, for which it is usually employed.

Next in line appeared the somewhat controversial duo-servo brake, often condemned for its excessive fade sensitivity, and yet being perhaps the most popular single brake in the U.S. The secret of this brake seems to lie in its taming, for on the stipulated basis it is superior to the 2L type, but it is only too easy so to design it that its torque factor is too high, when, of course, any other brake would become unstable too. The easiest method of taming is, of course, using a stable low friction lining which, if coupled with an efficient hook-up, leaves only one inherent deficiency, namely unequal work distribution between the shoes that usually consumes the extra fade

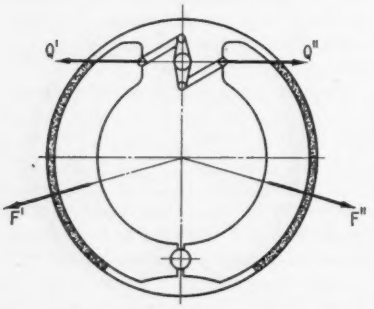


Fig. 15.

stability. Admittedly, a fade of the primary will always fade the secondary shoe too, but in practice the shoes of a 2L brake have the habit of fading together, except in some highly improbable conditions, so that this point seems to border on the hypothetical.

By way of a positive feed back, it is possible to use a part of the anchor reaction for augmenting the externally imposed shoe tip load. Ultimately, however, nothing seems to be gained functionally over an ordinary leading shoe, whatever complications are introduced. The ordinary LT. brake seems rather inferior compared with the 2L type, in everything except simplicity of construction. Its great fade sensitivity will be seen when remembering that a single shoe, with little help, has to do the work of two. This calls for about 50% increase in shoe factor to bring it level with a comparable 2L brake.

Reference has been made to the physiological aspects of fade and pedal load, the arguments having been based on the validity of the Weber-Fechner law. The author's tests were obtained, however, in a "quick check" manner, which was quite satisfactory for some practical purposes, but left many questions unanswered. For instance, one of the interesting points is the fact that the sensation of pedal effort seems to depend on the sponginess of the pedal. Most people tend to push a rigid pedal about twice as hard as one with about 8" movement. It would thus seem probable that an optimum pedal layout could be found analogously to the well known optimum size of hand-cranks (Hütte, 1929). A plea is made therefore, that this subject should be further investigated by Universities or other Research Institutions.

As mentioned these notes attempted to assess the reaction of various brakes to a certain friction fade, and did not take into account that friction fade itself considerably depends on the type of brake. In particular, it will be obvious that the more uniform the lining loading is, the less fade will ensue. Consequently, the ideal brake should have a very large number of very small shoes, each equally loaded so that the effective and nominal contact areas should be more nearly

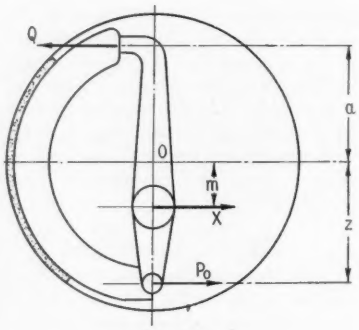


Fig. 14.

the same. Conversely, it would follow that, for the same duty, such a brake could be made smaller, etc. With these reservations in mind, the true purpose of these notes could be said to lie in clarifying perhaps a few ideas, and reorientating some prejudices. Finally the author wishes to acknowledge indebtedness to the Directors of Girling Ltd. for facilities and permission to publish these notes, in the compilation of which Mr. J. H. Pratt, M.I.Mech.E., not only took a great interest, but gave some invaluable advice, too. The author has to thank Mr. M. K. Booth, B.Eng., C.I.Mech.E. for much of the tedious work done.

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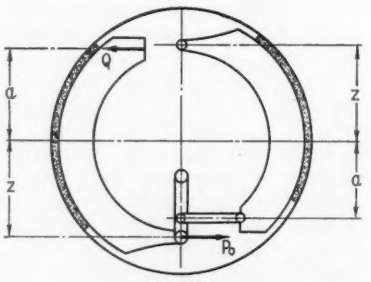


Fig. 17.

Appendix I

ANALYSIS OF FIXED ANCHOR SHOES

After deriving fade sensitivity, the main functional equation of fixed anchor shoes is established connecting shoe factor, fade sensitivity, shoe tip travel, lining arc, wear pattern with lining index that gives the frictional value of a lining. This is followed by some further corollaries.

I (a) Main Equations

Fig. 16 illustrates diagrammatically a fixed anchor shoe with the centre of drum drag having radius vector (L), argument (θ) and friction angle (ρ). From geometry it follows then that:

$$\Psi = \rho + \theta - \frac{\pi}{2} \quad (1.1)$$

where $\mu = \tan \rho$ (1.2)

The torque (T) is then

$$T = F.L. \sin \rho \quad (1.3)$$

and the shoe factor (K) is

$$K = T/Q.R. = (F.L/Q.R.). \sin \rho \quad (1.4)$$

Now the arm (z') of (F) about the anchor is

$$z' = z. \cos \Psi - L. \sin \rho \quad (1.5)$$

so that on taking moments about the anchor

$$Q\,(a + z) = F\,(z.\cos\Psi - L.\sin\rho) \quad (1.6)$$

which on substitution into Equ. (1.4) results in

$$K = \frac{a + z.}{R} \cdot \frac{L.\sin\rho}{z.\cos\Psi - L.\sin\rho} = \frac{(1 + a/z)}{R.\sin.\theta/L.\mu + R.\cos\theta/L - K/z} \quad (1.7)$$

(N.B. This is valid for a leading shoe; for a trailing shoe the sign of μ changes.)

Referring again to Equation (1.7), it can be written as

$$K = \left|\frac{A\,\mu}{B - \mu C}\right| = \left|\frac{A/C}{B/\mu C - 1}\right| \quad (1.8)$$

(μ) being positive for a leading, negative for a trailing shoe, and

$$A = 1 + a/z = \text{(shoe tip travel) / (shoe centre lift)}$$
$$B = (R/L\,.\,\sin\theta \quad (1.9)$$
$$C = (R/z) - (R/L)\,.\,\cos\theta = (R/z) - B.\cot\theta$$

By a suitable choice of A, B, C and μ any value of (K) may be thus obtained, but with reference to Equation 1.8, it will be seen that only 3 of the 4 parameters are independent.

The value of (L) can be obtained for a sinusoidal pressure pattern—(Acres 1946).

$$L = \frac{4.R.\sin\tfrac{1}{2}\lambda\,.\sin(u - \theta)}{\lambda + \sin\lambda} \quad (1.10)$$

and

$$\cot(u - \theta) = \tan u.\,(\lambda - \sin\lambda)\,/\,(\lambda + \sin\lambda)$$

where λ = lining arc

u = angle subtended by bisector of λ and OX

I (b) Sensitivity and Fade

When (μ) changes to ($\mu + \Delta\mu$), (K) will change to (K + ΔK). On writing then

$$f = \Delta\mu/\mu \quad (1.11)$$

the fade ($\bar{S}$) is then defined as

$$\bar{S} = \Delta T/T = \Delta K/K \quad (1.12)$$

and its value can be easily found with reference to Equ. (1.8)

$$\bar{S} = \frac{f}{1 - (1+f).\mu.C/B} = \frac{(\mu + \Delta\mu).A}{\mu.A} \cdot \frac{B - \mu.C}{B - (\mu + \Delta\mu).C} - 1 \quad (1.13)$$

The sensitivity (S) may be now defined as

$$S = \lim_{f \to 0}(\bar{S}/f) = \frac{1}{1 - \mu.C/B} = K.B/\mu.A. \quad (1.14)$$

Also since the shoe locks for

$$\mu = \mu' = B/C \quad (1.15)$$

Equs. 1.13 and 1.14 may be re-written as

$$\bar{S} = \frac{f}{1 - (1 + f)\,.\,\mu/\mu'} \text{ and } S = \frac{1}{1 - \mu/\mu'} \quad (1.16)$$

This is valid for a leading shoe with ($\Delta\mu > O$); for a trailing shoe the sign of (μ), and for ($\Delta\mu < O$) the sign of (f), have to be changed.

I (c) Some corollaries:

(i) *The functional equation* of a shoe can be obtained by eliminating (μC/B) between Equs. 1.8 and 1.13

$$\mu C/B = 1 - \mu.A/B.K. = \bar{S}\,(-f)\,/\,.(\bar{S}1 + f) \quad (1.17)$$

resulting in

$$K = \mu\,.\,(1 + 1/f)\,.\,\frac{\bar{S}}{1 + \bar{S}} \cdot \frac{A}{B} \quad (1.18)$$

This is again valid for $\Delta\mu > O$ and for a leading shoe, the appropriate change in sign being necessary for other cases. With reference to Equ. 1.9, it will be also seen that

$$A = \text{(shoe tip travel) / (shoe centre lift)}$$
$$B = \text{wear pattern } (\theta) + \text{lining arc (L)}$$
$$f = \text{relative change in } \mu$$
$$\mu = \text{nominal friction}$$
$$\bar{S} = \text{actual value of relative torque change}$$
$$K = \text{nominal shoe factor}$$

Hence, other things being equal

$$K \propto \phi = \mu\,.\,|\,1 \pm 1/f\,| \quad (1.19)$$

ϕ being defined as the lining index, with its nomogram on Fig. 3.

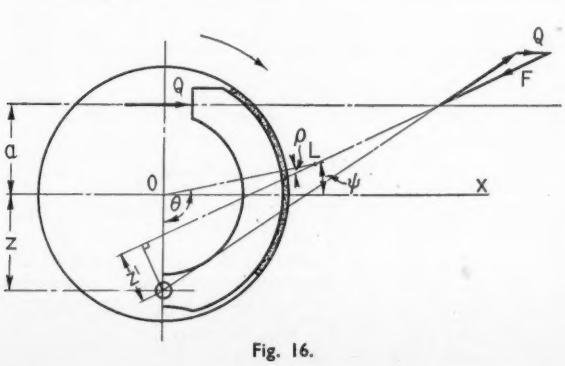


Fig. 16.

(ii) *Plotting shoe factor graphs.* Equ. (1.8) is evidently a hyperbola and in its inverse form, viz:

$$1/K = B/A.\mu - C/A \quad (1.20)$$

it is linear in (^{1}/K) and ($^1/\mu$), this being the basis for the graph paper on Fig. 7. Again, since Equ. 1.14 is also a hyperbola, in its reciprocal form, viz:

$$1/S = 1 - \mu.C/B \quad (1.21)$$

it is also linear in (^{1}/S) and (μ), forming the basis for Fig. 8. It will be also seen that this linear scale applies to single shoes only, and the torque factor of a brake mounting different shoes will, in general, not be linear. The deviation is, however, little, and hence accuracy is improved by using the same inverse and semi-inverse co-ordinates respectively. It also follows that if, in general, the torque factor curve is given by

$$1/Ko = f\,(1/\mu) \quad (1.22)$$
$$\text{then } dKo/Ko^2 = f'\,(1/\mu).\,d\mu/\mu^2 \quad (1.23)$$
$$\text{and } \therefore\ So = (dKo/Ko)/(d\mu/\mu) = (K/\mu).\,f'\,(1/\mu) \quad (1.24)$$

where the accent denotes differentiation. Consequently, by a process similar to that on Fig. 2, the sensitivity graph may be constructed more accurately than from an ordinary graph paper.

(iii) Servo gain

The increase in pressure between shoe and drum that is due to servo action can be found with reference to Equ. (1.6) from which

$$\frac{F}{Q} = \frac{a + z}{z.\sin(\theta + \rho) - z.\sin.\rho} \quad (1.25)$$

when ρ = O and F = Fo

$$Fo/Q = (a + z)\,/\,z.\sin\theta \quad (1.26)$$

so that

$$\frac{F}{Fo} = \frac{z.\sin\theta}{z.\sin(\theta+\rho) - L.\sin\rho} = \frac{K.B}{\mu.A}\sqrt{1+\mu^2} = S.\sqrt{1 + \mu^2} \quad (1.27)$$

from Equ. (1.14)

Appendix II

FADE OF BRAKE INSTALLATIONS

Hook-up and different shoe factors tend to weight individual shoe fade factors that can be compounded from details by means of general addition formulae which refer to two main cases: (i) to find the fade ($\bar{S}o$) of a single brake having two different shoes, and (ii) to find the fade ($\bar{S}oo$) of a complete installation comprising two different brakes.

Case (i):

Let it be supposed that a brake comprising two shoes, with shoe factors (K′) and (K″), fade factors ($\bar{S}'$) and ($\bar{S}''$) is acutated by load (W) acting through mechanical advantage (m). If the expander distributes the shoe tip loads equally, the torque (T) is given by

$$T = m.W.R.\,(K' + K'') = m.W.R.Ko \quad (2.1)$$

the torque factor (Ko) being defined for (n) shoes as

$$Ko = \sum_1^n K \quad (2.2)$$

For a certain change ($\Delta\mu$) of (μ), the torque will be then

$$T + \Delta T = W.m.R.\,(K' + \Delta K' + K'' + \Delta K'') \quad (2.3)$$

The fade ($\bar{S}o$) being defined as

$$\bar{S}o = \frac{\Delta T}{T} = \frac{\Delta K' + \Delta K''}{K' + K''} \quad (2.4)$$

and since $\bar{S}' = \Delta K'/K'$, etc. it follows that

$$\bar{S}o = \frac{\bar{S}'.K' + \bar{S}''.K''}{K' + K''} \text{ or } \bar{S}o\,.\,Ko = \bar{S}'.K' + \bar{S}''.K'' \quad (2.5)$$

Obviously if K′ = K″, $\bar{S}o = \frac{1}{2}(\bar{S}' + \bar{S}'')$, and neither the pedal load nor the mechanical advantage enters the picture.

Case (ii):

Let it be supposed that the brake installation of a vehicle is made up of two types of brake, characterised by torque factors (Ko′), (Ko″) and fade factors ($\bar{S}o'$) and ($\bar{S}o''$) actuated by pedal load (W) through leverages (m′) and (m″) respectively. If the brakes have radii (R′) and (R″), the torque is

$$T = W.\,(m'\,.\,R'\,.\,Ko' + m''\,.\,R''\,.\,Ko'') \quad (2.6)$$

If now there is some change in (μ) causing a change in (K), similarly as before

$$\bar{S}oo = \frac{\Delta T}{T} = \frac{m'.R'.\Delta Ko' + m''.R''.\Delta Ko''}{m'.Ko'.R' + m''.Ko''.R''} \quad (2.7)$$

and since $\Delta Ko/Ko = \bar{S}o$, etc.

$$\bar{S}oo = \frac{m'.R'.Ko'.\bar{S}o' + m''.R''.Ko''.\bar{S}o''}{m'.Ko'.R' + m''.Ko''.R''} \quad (2.8)$$

Formulae (2.5) and (2.8) are valid quite generally, but if sensitivity (S) is substituted for fade ($\bar{S}$), it entails assuming the same (f) for each shoe.

Appendix III

ON SOME SELF SERVO BRAKES

Ordinary leading and trailing shoes may be arranged to give an additional servo effect, generated through their positioning alone. Three such cases will be investigated below concerning duo-servo, self-servo and fixed cam brakes.

III (a) Duo servo brakes

Fig. 17 shows diagrammatically a generalised arrangement; an (n)th part of the primary anchor reaction actuates the secondary shoe, both shoes being otherwise identical.

Now the component (Po) of the anchor reaction can be obtained by equating moments about (O).

$$T = KQR = Po.z - Q.a \quad (3.1)$$

$$Po = (K.Q.R. + Q.a) / z \quad (3.2)$$

As (K) is the shoe factor of the secondary shoe too, and as its actuating force is (n.Po), the total torque supplied is

$$T = Q.K.R. + K.n.Po.R$$

$$\therefore \quad Ko = K.(1+n.(K.R.+a)/z) = K.N. \quad (3.3)$$

where $N = 1 + n.(K.R.+a)/z$ (3.4)

To obtain the overall sensitivity (So) the logarithm of Equ. (3.3) will be differentiated

$$\frac{dKo}{Ko} = So.f = \frac{dK}{K}\left[1 + n.K.R/z.N\right] \quad (3.5)$$

and with $dK/K = S.f$

$$So = S(1+n.K.R/z.N) = S\,[\,2 - (1+na/z)/N\,] \quad (3.6)$$

i.e. $S < So < 2S$.

It will be also seen that if (c = shoe tip travel of one shoe), the total shoe tip travel (Co) will be

$$Co = C.(1 + n.a/z) \quad (3.7)$$

III (b) Self servo shoe

A simplified version of this brake is shown on Fig. 14, the feed-back being n.Po, so that the total actuating force is

$$Q' = Q + n.Po \quad (3.8$$

Again by taking couples about O

$$T = K.Q'.R = m.X - Q.a = Ko.Q.R. \quad (3.9)$$

If then n = lever ratio

$$\therefore \quad m = (z - n.a) / (n + 1)$$

and $X = Po\,(1 + n)$

so that $R.K.\,(Q + n.Po) = Po\,(z - n.a) - Q.a$ (3.10)

i.e. $$Po = \frac{Q.(a+KR)}{z - n.\,(a + KR)}$$

which on substitution into Equation (3.8) yields

$$Ko = K.Z./[\,Z - n\,(a + K.R)\,] = K.M \quad (3.11)$$

where

$$M = \frac{1}{1 - n(a + K.R.)/Z} \quad (3.12)$$

To obtain the sensitivity (So) the logarithm of Equ. (3.11) will be differentiated

$$\frac{d.Ko}{Ko} = So.f = \frac{dK}{K}\left[1 + M.n.K.R/z\right] \quad (3.13)$$

$$\therefore \quad So = S(1+M.n.K.R/z) = S.M.(1 - n.a/z) < S.M. \quad (3.14)$$

The corresponding increase in shoe tip travel can be derived by considering that if

shoe tip travel with a fixed anchor = C
the anchor is moved by = n.C
which results in a further shoe tip travel = n.C.a/z
that causes a further anchor movement = n^2.C.a/z
increasing shoe tip travel by = C.$(na/z)^2$

$\therefore$ total shoe tip travel

$$= Co = C\,[\,1 + (n.a/z) + (na/z)^2 + \ldots.] = C\,/\,(1 - n.a/z) \quad (3.15)$$

III (c) Fixed cam brake

Fig. 15 illustrates the type diagrammatically, the fixed cam imposing a constraint that results in equal shoe tip movement for two otherwise equal shoes. On the assumption that these shoe tip movements obey Hooke's law, the forces between shoe and drum must be equal, i.e.

$$F' = F'' \quad (3.16)$$

It follows then that since

$$T = F.\mu.L \quad (3.17)$$

$\therefore$ K'.Q' =. K". Q"

where (K') and (K") are the individual shoe factors. The total torque (T) is thus

$$T = R(K'.Q' + K''Q'') = 2R.K'.Q' \quad (3.18)$$

but the total input load (Q) is

$$Q = Q' + Q'' = Q'\,(1 + K'/K'') \quad (3.19)$$

and the torque factor (Ko) is hence on equal shoe tip load basis

$$Ko = \frac{T}{\frac{1}{2}Q.R} = \frac{4.K'}{1+K'/K''} = \frac{4.K'.K''}{K'+K''} \quad (3.20)$$

Since both shoes are identical, with reference to Equ. (1.8), (3.20) may be written as

$$Ko = \frac{4.\mu^2.A^2.}{B^2 - \mu^2.C^2}\left\{\frac{\mu.A}{B + \mu C} + \frac{\mu.A}{B - \mu.C}\right\}^{-1} = 2\mu A/B \quad (3.21)$$

from which it follows that the overall sensitivity (So)

$$So = (dK/K) / (d\mu/\mu) = 1 \quad (3.22)$$

It will be also seen that the maximum value (Ko) can attain, if $K'' \to \infty$, is

$$\lim_{K'' \to \infty} Ko = 4.K' \quad (3.23)$$

Appendix IV

ON THE STABILITY OF BRAKES

A comparison will be made below between 2L and LT, DS, SS and NS brakes on the basis of common Ko, L, θ, μ and A, the criterion of superiority being the relative values of (So), assuming the same (f).

IV (a) LT. brake

On using two identical shoes with shoe factors (K'), (K") and sensitivities (S') and (S"), with reference to Equ. (1.14)

$$S' = K'.B/\mu.A \text{ and } S'' = K''.B/\mu.A \quad (4.1)$$

which on substitution into Equ. (2.5) result in

$$So = (B/\mu A)\,(K'^2 + K''^2) / (K' + K'') \quad (4.2)$$

If now a 2L brake is to have the same torque factor (Ko), its shoe factor (K) would be $K = \frac{1}{2}(K' + K'')$ and its sensitivity

$$So' = S = \tfrac{1}{2}(K' + K'').B/\mu.A \quad (4.3)$$

Now (4.4)

$$S/So = \tfrac{1}{2}(K'+K'')^2/(K'^2+K''^2) = \tfrac{1}{2}\,[1+2.K'.K''/(K'^2+K''^2)\,]$$

showing that if the LT brake is to be more stable

$$S/So > 1, \text{ i.e., } 2.K'.K'' > K'^2 + K''^2 \quad (4.5)$$

which is clearly impossible.

IV (b) DS. brake

It was shown in Appendix III (a) ante, that for a DS. brake

$$Ko = K.N \quad (3.3)$$

$$So = S.\,[2-(1+na/z)/N] = (K.B/\mu.A)\,[2-(1+na/z)/N.] \quad (3.6)$$

$$Co/C = 1 + n.a/z \quad (3.7)$$

If now a 2L brake is to have the same torque factor, its shoe factor (K') has to be

$$K' = \tfrac{1}{2}\,K.N \quad (4.6)$$

Its sensitivity (S') would be then with reference to Equ. (1.14)

$$So' = S' = \tfrac{1}{2}\,K.N.B/\mu.A' \quad (4.7)$$

If equal shoe tip movement is stipulated, then

$$A' = A.\tfrac{1}{2}(1 + n.a/z) \quad (4.8)$$

so that

$$So' = S' = (K.B/\mu.A).\,(N/(1 + n.a/z) \quad (4.9)$$

The DS. brake is thus superior if

$$So=(K.B/.\mu A)\,[\,2-(1+n.a/z)/N\,] < (K.B/\mu.A).N/(1+n.a/z) = So' \quad (4.10)$$

Since this simplifies to the condition of

$$O < (N - 1 - n.a/z)^2 \quad (4.11)$$

it follows that the superiority of the DS. brake is always ensured. The influence of various design factors can be then demonstrated by writing

$$y = (1 + n.a/z)/N = \frac{1}{1 + \frac{n.K.R.}{z + n.a.}} \quad (4.12)$$

$$So/So' = 2y - y^2 \quad (4.13)$$

the design improving as $y \to o.$, i.e. N and $K \to \infty$

IV (c) SS. brake

Referring to Appendix III (b) ante, it was shown that for a SS. brake

$$Ko = K.M. \quad (3.11)$$

$$So = S.M.\,(1 - n.a/z) \quad (3.14)$$

and $$Co/C = 1/(1 - n.a/z) \quad (3.15)$$

If now a 2L brake is to have the same shoe factor (K'), i.e.

$$K' = Ko = K.M. \quad (4.14)$$

its sensitivity (S') will be

$$S' = K.M.B/\mu.A' \quad (4.15)$$

If it is then further stipulated that the shoe tip travels should be also the same,

$$A' = A/(1 - n.a/z) \quad (4.16)$$

$$\therefore \quad S' = (K.B/\mu.A).\ M.\ (1 - n.a/z) \quad (4.17)$$

so that it follows that $So = S'$ (4.18)

IV (d) NS. brake

It has been demonstrated in Appendix III (c) that for NS. brakes

$$So = 1$$

their torque factor being

$$Ko = 2.\mu.A/B \quad (3.21)$$

If now a 2L brake is to have the same torque factor, and with A, B, etc. the same, its shoe factor (K') would be

$$K' = \mu.A/B$$

and its sensitivity would be $S = K'B/\mu A = 1$

Again this is only possible if C = O, i.e. the shoes have no servo action. In practice, this results in $z \to \infty$ and hence if $A \approx 2$, this would be possible only if $a \to \infty$. Therefore, the comparison is valid in reality only if A = 1, or a 2 : 1 lever, or like, is embodied in the brake.

DIESEL ENGINES

Some Notes for Manufacturers Exporting to Asian Countries

These notes have been compiled by an engineer having a wide experience of Eastern conditions in road transport. They constitute a summary of the points that have proved to be of chief importance in the maintenance of a large fleet of vehicles in an Asian country.

TO-DAY the East, Alexandria to Singapore, is gradually becoming diesel engine minded. There are however, certain factors that work against their extended use in these areas. These include higher first costs and landing charges, costly and mysterious accessories accompanied by instruction books that carry dire warnings, or the theme "return it to the makers" should anything go wrong or be neglected. Struggling say, over the mountainous regions between Afghanistan and Pakistan in shade temperatures ranging from 10° in winter to 115° in summer, carrying the fuel or fruit trades, it is poor consolation for a contractor to read "send it to the nearest service station" which means Bombay 1000 miles away, and where a delay of one month is not unusual while the part required is "aired" from the U.K. or another service station 1000 more miles away. The salesman's reply to this is carry duplicate accessories, but few would dare to suggest this to a petrol lorry owner. Axial self-starters and fuel injection pumps present real difficulties, and check sales of diesel units. What seems to be required is the anticipation of the neglect and abuse elements in their design and construction. Neglect and abuse they will assuredly receive in these areas.

Axial Starters

In skilled and diesel minded hands with considerable appreciation of the limitations of battery and starter, little trouble is found. There are however, many slovenly minded operators who overlook extremes of temperature, the defective condition of an engine, or the decompressor if fitted. When they have completely discharged the battery, they think nothing of bringing in series another and so double the voltage in order to get faster flywheel speeds. The results are buckled plates, burnt out starter windings and what is more disastrous, towing to start up and constant idling of the engine during rest throughout the day and sometimes far into the night.

Here the main obstacle seems to be the necessity for providing the cheapest voltage system. An increase from 12 to 24 volts would put many diesel engines out of the Asian markets, yet to cope with abuse, evidence indicates that a higher voltage is one way of preventing the early demise of the battery and self-starter. Probably the better way is to follow the lead of land and agricultural engineers and fit small, two-stroke petrol engine starters, and thus be free of batteries, expensive dynamos and the complicated axial starters. Dispensing with this starter would be one item less to send to the nearest service station; for it eliminates

(1) its intricate and delicate clutch which requires great care in servicing in order to prevent damage to the armature windings,

(2) the heavy wear of bronze pinion and its whitemetal bearing which causes heavy wear of flywheel teeth and

(3) the rapid destruction of the battery by abuse.

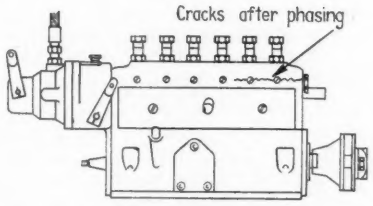


Fig. 1. Location of cracks in pump casing.

All these factors operate against the success of axial starters in Asian countries where for some time to come there is little hope of improvement in servicing these parts.

Nothing is more simple to maintain than the battery upon which the success of the starter rests, yet no item is more neglected in Asian countries. The battery is out of sight and out of mind, for few seem to bother about greasing of terminals, topping up, checking of specific gravity, or even extra charging to cover excessive discharging in winter, when starting loads can be very severe.

Fortunately there are to-day two reliable alternatives to electric starters: (a) small petrol engines now found on American land machines, and (b) the recently introduced hydraulic starters to British diesels. If the latter prove as reliable as hydraulic brakes, their eventual replacement of electric starters will be welcome to many overseas operators.

Fuel Injection Pumps

Despite the remarkable progress made in their design and manufacture during the last ten years, they are still as much a sealed mystery as was the magneto before 1918, when followed a change from rotary armatures to stationary, which removed the mystery and simplified maintenance. Some such revolution in design seems necessary in order to free the fuel injection pump from costly service stations and the mystery with which it is now enshrouded. At present it has inherent weaknesses of which the following seem the most important to overcome in order to increase its life between repair periods:—

(a) The output of each plunger depends upon accurate re-calibration after certain periods of service and depending on the amount of element wear, which requires a highly skilled staff and delicate equipment.

(b) The phase angle of the pump depends upon maintaining correct adjustment of the operating tappets, and the method advised by the makers can and does cause split pump casings where ham-fistedness is the rule. See Fig. 1.

(c) Filtering elements provided outside the pump are insufficient to prevent premature wear of pump elements by dirt and moisture in the fuel.

(d) The type of leather diaphragm employed in the pneumatic governor is unreliable and liable to permit excessive r.p.m., which if not detected can wreck the engine.

(a) Calibration is likely to remain a much neglected subject in Asian countries. Lack of skill, knowledge, and delicacy of touch, fragile equipment with its test tubes and method of hand turning, all inevitably lead to errors in calibration. The writer prefers a home-made bench test at 700 and 200 r.p.m. with the control rod set at 12 m/m and 6 m/m, the output of each element being connected to special pressure-set injectors mounted against

a graphlike background so that each "squish" could be sized and sounded, any over or under being rectified in the prescribed manner until all were the same size and sound. Fig. 2 illustrates this idea.

(b) Phasing the pump is one of those operations which magneto experts find it hard to believe is necessary. To translate the makers' instruction books on this delicate operation in the many Asian languages and dialects would be no light task for a linguist, let alone the diesel engineer. In one make 15 operations are necessary. In one, in order to determine the plunger clearance, operators are instructed to "turn the camshaft until No. 1 is at top of its cam, and then insert a screwdriver between the tappet adjusting screw and its locknut; by gentle leverage the plunger assembly is lifted from its cam and the plunger is pressed against the underside of the valve seat".

The ham-fisted operators lever the plunger hard against the valve head and set up internal stresses in the pump casing at its weakest section. Cracks as in Fig. 1 appear soon after phasing operations have been carried out, and other cases appear some little time after with the help of a blocked injector, a common fault with engines operating for long period at extreme loads. Levering up the plunger to find its clearance is not likely to help Asian operators, so another though less dangerous method is advised. "Carefully raise the tappet adjusting screw until the clearance is nil, i.e., the plunger is touching the underside of the valve seat", and then "lower the plunger by giving the screw half a turn back to get the required clearance of 0.020".

This operation is certainly dangerous enough in a world where brute force and ignorance are the rule. It would appear safer for the maker's reputation to avoid the use of screwdrivers and simplify the instructions for phasing by sketches, and also illustrate a cracked casing due to ham-fistedness. But why cannot the plungers be crank driven and the pump as free of phasing as a magneto? It should not be difficult in these mechanically enlightened days.

(c) Having experienced 200,000 miles of service from polar-inductor type magnetos in a desertlike climate, the writer was disappointed with 15,000-50,000 miles from fuel injection pumps, which then needed new elements. More resistance to wear by better filtration than at present stipulated or provided seems desirable. Few could question the HSD fuel supplied in this area by the Shell organisation, which has :—

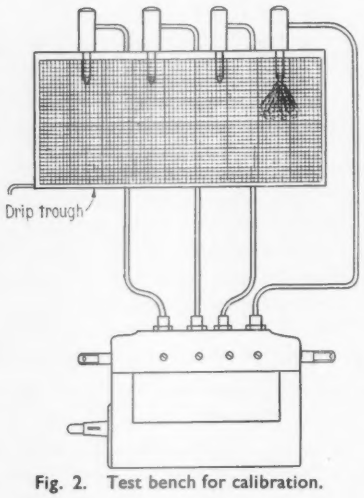


Fig. 2. Test bench for calibration.

0.8390 spg. at 60°F.,
190° flash point,
0.7 per cent. sulphur (some would like to see this lower),
Negligible carbon residue,
Nil water, ash and asphaltenes,
60.5 diesel index,
56 cetane No.,
19500 B.T.U. calorific value.

However, makers would be well advised to warn land users in remote regions against the use of furnace oils. The writer has seen these used with disastrous results to big ends, cylinder liners and elements, as well as the maker's reputation.

Much unnecessary element wear could be stopped if fuel received from installations could be stored in such a way as to be free of contamination from moisture, dust and even rain. Probably the best system to employ for small and medium sized fleets is to have one underground receiving tank supplying by pump to overhead settling and delivery tanks, each overhead tank having three days for settling. Gravity filling through perfect filters is much superior to pump filling from underground tanks, since the pump agitates the settlings. And it is at the filling station that much if not all the sediment and moisture could be filtered and so save the small filters on the units. These are of zinc base alloys and do not stand up to the frequent handling recommended. Threads stretch, there are constant leaks and collection of diesel grime occurs. Pre-filling filtration is therefore advisable at the filling stations where the task of trapping grit and moisture is easier.

A new approach to this problem seems advisable. In Asian countries as in the West, labour is less amenable to routine tasks, therefore filtration should be placed on a self-cleaning basis. The fuel distribution companies should help in the introduction of pre-filling filtration at the filling stations, while makers should revise their instructions on the subject of cleaning filters. More harm has been done to pump elements by regular cleaning of filter packs on the vehicle than by completely neglecting them. Opening filters invariably means grit and fine sand getting in, particularly if a sand storm is raging, and with labour as it is to-day, it is next to impossible to obtain the scrupulous cleanliness that these filters demand. The cloth pack filter can be neglected for one year and still be capable of filtering, while the felt type can give two years' service. It would seem better for makers to specify that the filters be left alone until they completely block the fuel, and then advise users to change them, or alternatively advise changing the cloth type every year and the felt every two years. But with pre-filling filtration at the filling stations the vehicle filters would have little to do.

(d) The leather type diaphragm employed in the pneumatic governor mechanism of the fuel pump is not always satisfactory in shade temperatures of 80 to 115°F. Split or perished leathers occur, and not always being detected have caused overrunning and wrecked engines. It seems the experimental stage with governors has yet to be passed. Weights gave place to pneumatic control, the leather diaphragm being lubricated with mineral oil (a lubricant no equestrian would recommend for leather), and this is now under threat of displacement by hydraulic control. It is hoped that success will soon be forthcoming, for more refined control rod motion than pneumatic governing can give, is necessary for smooth acceleration and safe limitation of engine speed. It should not be necessary to warn newcomers of the possibility of wrecked engines if the leather splits or the venturi pipe has been finger-tightened, as frequent a defect in this part of the world as ham-fistedness.

A fuel injection equipment capable of the maximum amount of abuse and neglect is therefore essential for Asian countries, equalling to-day's ignition equipment on petrol engines.

Lubricating Oils

It is not always easy to select the right oil. Temperatures, humidity, light, heavy or severe operating conditions, dust, and at the coastal regions, salinity, all can upset the selection of experts. The writer was once advised by a worldwide oil company's chief chemist in London to use a heavy detergent oil for temperatures of 105°F., shade. This caused so much

heavy sludge that several engines were wrecked through collapsed big ends and worn timing chains altering the timing and permitting the valve heads to strike the piston crowns. This same expert recommended S.A.E.20 oils for petrol engines in temperatures of 85°F. much to the horror of his local colleagues. The advice of experts with no knowledge of local conditions is seldom sound, and test reports seldom agree with those from far off regions. To confuse matters further, excellent reports on HD oils from one continent are opposed by bad reports from another, for the reasons mentioned.

Before it is safe to recommend a particular lubricating oil it would make the selection easier for makers if they had used oils laboratory tested and the results plotted for the various countries and climatic conditions under which their engines operate. The writer feels sure the oil companies would help in this valuable work. Oils used in the hottest as well as the coldest times of the year should be tested after 5000 miles use, the user being instructed to see that the oil filter is regularly cleaned. With the percentage of total solids further divided into percentage of iron, silica, copper and ash, coupled with the percentage increase in viscosity, selecting lubricating oils for diesel engines exported to Asian countries would be simplified. Makers would also be able to see where harmful operational faults existed, and their engine's own shortcomings; also of equal importance the efficiency of the oil filter provided.

When to change oils is another problem. Some manufacturers are too fond of specifying every 2000 miles. 5000 miles is perfectly satisfactory and fits in with other problems. Above this the viscosity increase is steep and harmful. Reliable oil filters readily cleanable are essential. Though ring sticking, port clogging, piston skirt and ring land deposits seem less than in temperate climes, cylinder wear is usually heavier. This does not in the writer's view call for the use of HD oils which carry the grit and carbonaceous matter round in suspension, are unfilterable and unstable. Straight mineral oils remove the fear of mysterious failures and drop all the carbon and grit on the bottom of the sump where it is harmless and out of the way. By virtue of being filterable, stable through lack of detergents that decrease with use, and having less corrosive elements, mineral oils must remain a better financial proposition. If there is any rise in the percentage of sulphur in the fuel of the future, would it not be sounder engineering practice to improve materials with the help of metallurgists rather than leave it to the scientists and their dopes?

Increasing engine speeds to raise the bhp has not always proved sound, there being no margin to deal with the effects of high altitudes, temperatures and loads. There is an economical engine speed for fuel, oil and replacements, generally round about 1600 rpm, and above this usually means more crankshaft and bearing troubles. The move towards maximum speeds of 1700 is therefore welcome. It will probably be some time before inertia loads of diesel engines permit them to compete with petrol engine speeds.

A crankcase extending well below the centre line and a sump, without oil rings, that has an inspection cover with a plug the threads of which will withstand frequent removal, are required. There will be rough driving on rough roads, so suspension brackets need to be at the very ends of the engine and gearbox, and their bolts equal in diameter to those employed for main bearing caps. Side projecting brackets are likely to loosen or fracture. Push fit cylinder liners are welcome to a land where many cylinder blocks are distorted removing liners.

Crankshafts

Crankshafts are still the weakest link. The heavy scoring of the impact arcs of the journals and crankpins make centre balance weights and nitrided hardening necessities. Both can extend the overhaul periods by 100 per cent. The writer's experience is that more unhardened shafts break across the webs due to heavy wear increasing the running clearances, than by fatigue or distortion arising from nitrided hardening, which some makers fear.

Hardened thrust washers are more positive against end-play than flanges of main bearings, particularly in the case of passenger vehicles where clutch operation is more frequent. This is important for oil economy, otherwise crankshaft end-play soon lets oil past the flywheel oilring. Oil spill at the flywheel is far more of a problem with diesels than with petrol engines due to cramped labyrinths. Here the addition of felt rings would help to prevent breathing and that oil blow-by that sometimes occurs with worn engines and makes them so extremely wasteful and costly in lubrication in comparison with petrol engines.

Ring sticking and varnish are not usual in hot climates, but cylinder wear is greater than in temperate climes, sometimes three to five times greater. Some laboratory tests might therefore be made in the cylinder group in order to determine the best combination of materials to employ. The writer favours hardened liners and piston lands, though emphasis must be on the rings, heavy ring wear usually resulting in heavy cylinder and land wear. Oil-bath air cleaners of ample capacity should be fitted, but one frequently finds filters on diesel engines the same size as those on 8 h.p. cars. Where temperatures are above 90°F., outside the bonnet in a cool position is the best place for the air filter. Where severe oil control rings are necessary, regard should be paid to cylinder wear rather than to fantastic oil economy which soon disappears through rapid wear. Lubricating oil requires no import license and is easier to replace than cylinder liners and pistons.

The valve assembly is not without its troubles. Chain driven camshafts through weak mounted chain tensioners and rapid wearing chain pins upset the valve timing and sometimes wreck the engine when the operator is not sensitive to the noise of the valve head hammering the piston crown. Driving by gear trains is more positive and safer. Hardened seat inserts are desirable, likewise valve seats and stems. Wear is much more rapid in the East than in temperate climes.

Water jackets should be as ample as possible and free of pockets in order to avoid water locks and cracks between the valve seats, for the water is invariably hard and prone to heavy scale. Detachable covers for descaling are necessary. Water softening is difficult to obtain and a monthly runout of water is seldom given. In fact running out the sludge is another much neglected item in this area. If thermo-syphon cooling can be successful on 8 h.p. cars in the tropics to-day, as it was many years ago, why is a water pump necessary on diesel engines where the waste heat is less? The modern water pump seems to get worse rather than better. Weak glands cause scouring of shafts or bearings, or vice versa, overheating due to shortage of water occurs and unnecessary wear is introduced, not only in the water pump but in the cylinder and valve groups. Complaints against the water pump invariably bring the reply that it is not being regularly greased with the correct grease and so this accessory continues its annoying way forcing refilling of radiators every few hours instead of days.

Adequate cooling seems just a question of a radiator of ample capacity with the maximum air stream. Many British "out in the open" type fans merely blow a weak air stream over the engine, whereas a fan of ample air flow cowled as close as possible to the radiator must permit more efficient cooling than to-day's water pumps, and would free maintenance engineers

of many petty repair problems. If this cannot be done, at least let us have pumps designed by hydraulic engineers and eliminate the jigsaw puzzles that exist to-day.

Conclusion

Instruction books are better for less technical jargon and more illustrations. They should be in simple English in order to be correctly understood. After sales it is worth while telling buyers what spare parts are necessary after certain periods of running. Import licenses can then be arranged in time to avoid idle units while spare parts are imported. The question of price is very important in lands where the standard of living is probably the lowest in the world. Exchange and import controls are severe obstacles to overcome, and with mounting Continental and American competition, and probably some to come from Japan, the spiral of rising British prices chasing rising wages and materials must end if Britain wishes to continue the present volume of automotive exports. Devaluation has afforded some advantages; probably others could be obtained by curtailing vices and the enormous taxation the British Government obtains therefrom. No other countries, except perhaps the U.S.A., seems to have such excess spending power for drink, gambling, entertainment and the travelling it entails, as British labour has to-day. To further demands for more wages some plain words need to be said otherwise the collapse of Britain's automotive exports to Asian countries must occur. Britain only exports on price, and every one pound off the F.O.B. price means 30 to 35 shillings less for the buyer to pay.

Finally there is the question of size and output. A good four cylinder diesel engine with a maximum rpm of 1700, capable of at least 30 bhp at 1000 rpm and 45 at 1500 rpm, is an ideal size for the replacement of petrol engines at present found in the cheap two and three ton trucks and buses that abound in Asian countries. Why have six cylinders with costlier accessories, spares, running and capital outlay ?

DRAWING AND HANDICRAFT COMPETITIONS

Brief Details of the I.B.C.A.M. Requirements

DETAILS of the 1951 series of Drawing and Handicraft Competitions arranged by the Institute of British Carriage and Automobile Manufacturers in association with the Society of Motor Manufacturers and Traders, the National Federation of Vehicle Trades and the Worshipful Company of Coachmakers and Coach Harness Makers, are now available and may be obtained on application to the Institute of British Carriage and Automobile Manufacturers, 50 Pall Mall, London, S.W.1.

Competition No. 1 is for a coloured drawing of a two-door coupe, scalé 1in. to the foot, in perspective and half plan. An outline drawing, scale 1½in. to the foot, is also required, together with a drawing to the same scale showing the hood in the closed position. The design is to be for a "one off" body, thereby giving full scope for styling and originality.

Competition No. 2 is for an outline drawing in ink of a four-door, four or six light saloon of metal construction suitable for mass production at a rate of 200 to 300 bodies per week. Full constructional details must be given of (a) scuttle and windscreen framing, (b) rear of body behind rear squab, (c) master sections for roof, pillars, doors, etc. A coachbuilder's chassis drawing will be supplied.

Competition No. 3 is for an outline drawing of an estate car suitable for submission to a prospective customer. A coachbuilder's chassis drawing will be supplied.

Competition No. 4 is for a general arrangement drawing of a luxury coach suitable for Continental touring, designed for mounting on a chassis with under-floor engine. Construction is to be either composite or all-metal, and a coachbuilder's chassis drawing will be supplied.

Competition No. 5 is for a general arrangement drawing of a mobile shop with provision for external and internal display of goods for sale as well as service facilities. Any suitable trade may be selected as the subject of the design. A coachbuilder's chassis drawing will be supplied.

Competition No. 6 is a handicraft competition for which the entries must be either a portion of a standing pillar and door, hung on concealed hinges, to a drawing to be supplied to competitors, or a hand beaten wing spat.

A first prize of £50 is offered for Competitions 1, 2 and 5, together with additional awards. For Competition No. 3 the first prize is £15, and for Competition No. 4, £75.

In the Handicraft Competition a sum of £40 has been allocated.

Competitions 1, 2, 4 and 5 are open to all persons of British nationality without age limit. Competition No. 3 is open only to students of any technical school or employees in British workshops in the United Kingdom who are under the age of 21. The Handicraft Competition is open to students of any technical school or any employees in British workshops in the United Kingdom under the age of 21 and to competitors under the age of 23 who have served not less than twelve months in H.M. Forces. The closing date for entries is May 5th, 1951.

Joint Engineering Conference

PROVISIONAL arrangements for the Joint Engineering Conference to be held in London from June 4th to 15th have recently been announced. It will be held at the Institution of Civil Engineers, The Institution of Mechanical Engineers and the Institution of Electrical Engineers.

Among the technical sessions of particular interest to automobile engineers are Road Transport (Mechanical Engineering) on Monday, June 11th, and Road Transport (Civil Engineering) on Tuesday, June 12th.

Papers will be presented at the Conference in abstract, so that the major portion of each session can be devoted to discussion. Advance copies of the papers will be available some time before the date of the Conference.

There will also be a programme of visits to Works and other places of interest, opportunities to visit the 1951 Exhibition and certain social functions.

Further particulars can be obtained from the Secretariat, Joint Engineering Conference, the Institution of Civil Engineers, Great George Street, London, S.W.1. (1932)

CHASSIS ASSEMBLY

Important Developments at the works of Leyland Motors Ltd.

DURING the past three years Leyland Motors Ltd., Leyland, Lancashire, have been carrying out development projects that will greatly increase the output capacity and efficiency of their factories. These have involved re-tooling and re-organizing the detail machine lines, a change in manufacturing methods from batch to flow production, and the introduction of assembly conveyors.

The first stage of the re-organization was the introduction of flow production instead of batch production in the machine shops. This in itself, was a major contribution to increased efficiency. With flow production, the time that elapses between the initial and final machining operations on a component is greatly reduced. As a result, the amount of work in progress is also greatly reduced, with a consequent reduction in the amount of capital locked up. A further advantage is that delivery dates are much more easily maintained.

Machine operators employed under this system work in groups and have a pooled bonus scheme based on the group output of completed parts. When necessary the operators move freely from one machine to another in the group. This is an important factor in maintaining an even flow of parts and in preventing an accumulation at any stage along the machining line.

The second stage of the project consisted in breaking down and replanning the various assembly units and locating them in the most appropriate positions for feeding the chassis assembly line. Wherever possible, the units are assembled on conveyors. The final stage of the project has been the installation of a conveyor, in three sections, for chassis assembly. This conveyor has a total length of almost 1000ft. It is believed to be the only moving conveyor of its kind on which heavy duty chassis are being erected.

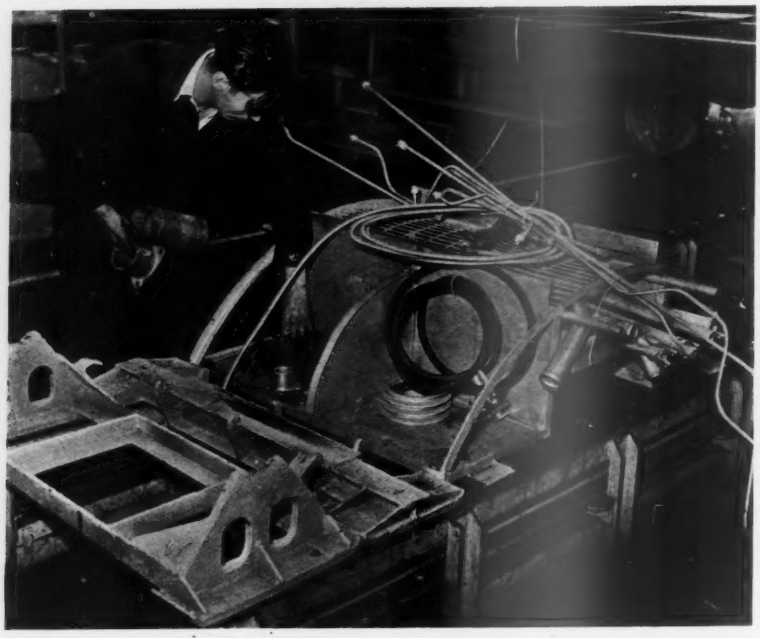
Production Control

Efficient assembly of heavy vehicle chassis on a moving conveyor is possible only if there is very close production control to ensure that the production of sub-assemblies conforms to the requirements on the conveyor. The system must be such that only those units are produced which will be used immediately for chassis assembly. This reduces the amount of work in progress at the sub-assembly stage, as the substitution of flow in place of batch production reduces the work in progress in the detail machining sections.

The basis of the production control system is the "daily conveyor rota". This shows exactly what chassis are to be laid down on the conveyor on a specific day. Each chassis is given a conveyor number that indicates its rotation on the conveyor line. Also listed on the rota are the sub-assemblies needed to build each chassis. When a sub-assembly is completed, it is marked for the same conveyor number as the chassis for which it is made. In brief, every sub-assembly is built for a specific chassis.

At the longest, the daily rota is compiled only five days ahead of the date at which chassis are to be laid down. In practice, the engine factory requires only 72 hours notice to build the type of engine that is required, and 48 hours notice is sufficient for the gearbox assembly. As the sub-assemblies are completed they are despatched direct to the appropriate section of the conveyor line. Fittings for each chassis are marshalled in the stores and placed in trays which are also marked with the conveyor numbers before being despatched to the assembly conveyor. These trays are eventually placed on the conveyor immediately behind the chassis for which they are to be used. Since they travel along with the chassis they are to hand and ready for use when required.

The chassis assembly department at Leyland Motors Ltd.



Small parts in marshalled sets travel along the conveyor immediately behind the chassis.

The Chassis Conveyor

There are three parallel lines of conveyor with respective effective lengths of 250ft., 350ft. and 300ft. Each line has two endless roller chains 3ft. apart at centres. The chains are of the offset link bush roller type with 8in. pitch links and 3in. diameter rollers. Offset links are employed to provide a continuous flat top run to carry the chassis.

Both top and bottom runs of each chain operate in continuous fabricated channel troughing. The tight side of the chain is on the top. Supports for the upper channel troughing are placed at 2ft. intervals so that the maximum chassis weight likely to be encountered can be carried safely. The bottom run of the chain is guarded and there are suitable guards for the chain sprockets.

All the driven wheels have 12 teeth and a 30.9in. pitch circle. The driving sprockets for the 300ft. leg have 8 teeth and a 20.9in. pitch circle, while the driving sprockets for the 350ft. and 250ft. legs have 12 teeth and 30.9in. pitch circle. Each driven wheel is carried in plummer-block type bearings mounted on a cast slide base. Correct chain tension is maintained by means of a screwed bar with a lock nut on each side of a fixed boss. Lubrication is provided by an oil bath under the sprocket, into which the slack run of the chain dips.

The primary drive for each conveyor is supplied by a Stamford squirrel-cage 5 h.p., 14/40 r.p.m. high torque electric induction motor. From the motor the drive is taken through a vee belt to a Croft variable speed pulley, thence to a Holroyd double reduction box and finally to a Holroyd single reduction unit. Each pair of driving sprockets is keyed to the output shaft of a final reduction unit, and the shaft is supported at its ends by outrigger brackets. A spare electric motor is kept in readiness. It is interchangeable with any of the three in use.

On the variable speed drive the normal ratio is 3 : 1. The total reduction from the motor to the output shaft is, however, so great that the section of the vee belt for the drive from the motor is only $1\frac{1}{2}$in. $\times$ $\frac{7}{16}$in. Each drive unit is sunk in a concrete lined pit and is mounted on a large cast iron base that is grouted to the floor of the pit.

Each conveyor is built to run at a maximum speed of 5ft. per minute, but the speed is steplessly variable between 2ft. 6in. per minute and the maximum. At present one of the lower speeds is used. Compressed air lines and cables for electric sockets for power tools run along the length of each conveyor. There are overhead cranes for feeding components to appropriate points, road wheels are fed to the correct section down an elevated runway, and along the side of the final leg there are high-pressure lubricating guns.

Assembly Method

Before any work is carried out on the conveyor there is a preliminary assembly on the shop floor, when the springs are attached to the axles and the axles to the chassis frame by the shackle pins. This preliminary assembly is then lifted so that the front and rear axles rest on two channel section supporting beams, or cradles, lying across the conveyor chains. At this stage, assembly on the conveyor begins. Towards the end of the first leg of the conveyor system there is an overhead gantry for feeding assembled engines

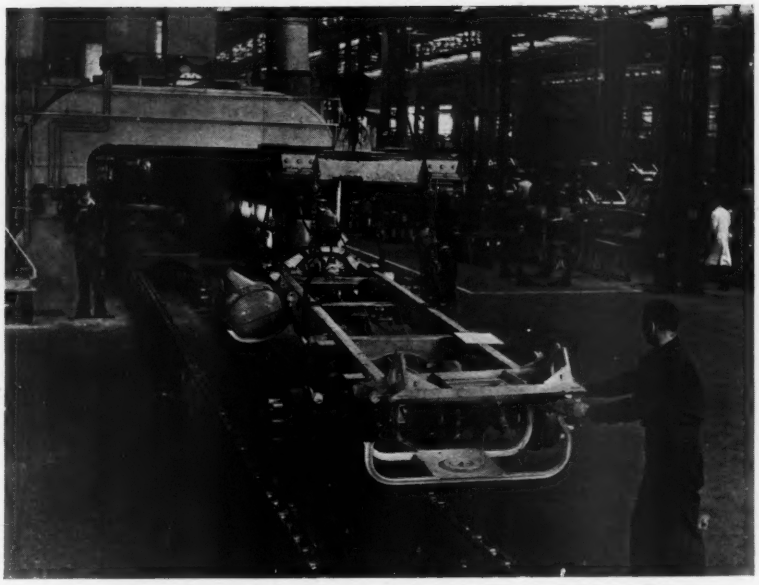
The end of the first line with a chassis lifted ready for transfer to the second line.

and gearboxes into the chassis. At the end of this leg, each chassis is lifted by crane and transferred to the second or middle conveyor line. When the work on the second leg is completed, the chassis is again lifted by crane and transferred to the third conveyor line.

As a chassis approaches the end of either the first or second lines, it trips a lever that sounds a warning bell to indicate that the chassis is ready for transfer to the next run of conveyor. Should the warning bell be disregarded, a second safety device comes into operation to bring the conveyor to a standstill. It is therefore, impossible for a chassis to overrun and drop off the end of the line.

Immediately a chassis is transferred from the second to the third run of conveyor, it is masked for paint spraying by four operators. When the masking is completed, the chassis enters a tunnel type spray booth, 20ft. long, 14ft. wide and 10ft. high. This booth is lighted by eight twin fluorescent tubes fitted along the walls. Air is extracted by two 30in. diameter axial flow type fans. These are mounted above the booth and draw air from the interior through removable wood wool filters to discharge via ducting through the factory roof.

The paint for spraying is stored in a 100 gallon pressurised container. Automatic paddles, rotating at 100 r.p.m., continuously stir the paint. There are two containers, and as one goes into use, the empty one is returned to the paint suppliers for re-filling. When the containers are being changed over, the air is cut off from the gun lines to prevent the paint from solidifying in the pipes.



The start of the third and final line showing the paint spray booth.

On leaving the spray booth, the chassis immediately enters a gas-heated drying oven. This has a total length of 44ft., including a 3ft. vestibule at each end. To prevent the escape of heat, the oven is sealed at each end by a curtain of cold air forced upwards at high velocity. The oven itself has automatic starting by means of sparking plug ignition. It incorporates a number of automatic safety devices so that in the event of circulating fan failure, the gas supply is cut off by an electric mercury switch. Other devices prevent any danger that the fans will cause circulation of unburnt coal gas. The normal working temperature for the incoming air is 245 deg. F. This is maintained by thermostatic controls. By the time the air circulates round the chassis, its temperature drops to about one-half of the original figure.

On leaving the drying oven, the chassis passes a battery of Tecalemit greasing towers and high-pressure lubrication guns suspended from swinging arms. These are for lubrication of the major units such as rear axles and gearboxes, and for general greasing. Finally, the cabs are mounted on the goods models and wheels with the correct tyre size for the type of chassis roll down a chute to be fitted. As the chassis is completed it trips a lever that rings a warning bell to summon the driver. At the end of the final leg the chassis rolls gently down a ramp to the shop floor, from which it is driven to the running shop and testing bay.

All types of Leyland four-wheel chassis are assembled on this conveyor system. This raised one of the most difficult of the planning problems. Obviously, some models will take longer to assemble than others. Nevertheless, the work must be arranged to give a balanced time factor at every stage. To solve this problem, the times taken at each stage for each different model were carefully worked out. Following this a system was planned whereby there would be a constant small pool of men who could be occupied either in sub-assembling parts for vehicles that take longest to

A completed Leyland Comet lorry leaving the final line.

build, or as additional hands on the conveyor itself at points where the work had to be speeded up to give a balanced time factor. In addition, No 2 conveyor was made longer than was necessary for the lighter vehicles in order to absorb the longer assembly time necessary for building the heavier chassis. The heavier vehicles travel the whole length of No. 2 line before transfer to No. 3 line, but the lighter are transferred before reaching the end of the line.

Basically, the number of men allotted to the conveyor is fixed, and they all have a fixed amount of work to do. The men of the pool even out differences in time factors. For example, the two or three hours extra work involved in assembling the pipe of air brakes on certain passenger models is done as a sub-assembly job by men from the pool, and the piping is then fitted as a unit with no loss of time in the general system. Payment of the men working on the assembly line is based on the output at the end of the conveyor and on the group piece-work principle. Every model has an independent and separately assessed piece-work allowance. It is found that this method of payment provides a definite incentive to increased production.

Increased production by simplifying assembly through a carefully planned operation sequence, and without imposing any additional strain on the operatives, is the object for which this conveyor system was installed. In fact, through the introduction of the conveyor, heavy manual labour has been greatly reduced, because the mechanised line brings the chassis automatically to certain fixed points associated with specific operations. It therefore lends itself to the introduction of power tools, many suspended from overhead runways and supports, and the provision of mechanical handling equipment at the correct points along the lines. Actually, this conveyor has a potential output of 10,000 chassis a year, twice that of the highest Leyland pre-war production figures.

Other advantages are also obtained from assembling chassis on a conveyor. Floor area can be used to greater advantage since there need be no large stocks of parts on the shop floor. In addition, so that there shall be an even flow of work along the conveyor, every department must work to a master production control plan. As a result, the work of the various departments is much more closely integrated. In brief, not only are total manufacturing times appreciably reduced, but in addition delivery dates can be specified with greater accuracy and the body builders and the road test department can be supplied with greater regularity.

That the project has been worth while and is successful has already been proved. The total cost of redeploying existing equipment and the purchase of new plant has been well over £1,000,000. Despite this large capital outlay, the greater efficiency has allowed the selling prices of Leyland vehicles to be stabilised at the 1947 level, by enabling the Company to absorb the increased costs of raw materials and bought-out parts, with the exception of recent tyre increases, without passing these on to customers.

SIZE EFFECT ON ENGINE DESIGN

ACCORDING to C. F. Taylor, in a paper entitled "Effect of Size on the Design and Performance of Internal-Combustion Engines" published in *A.S.M.E. Trans.*, Vol. 72, No. 5, many factors in determining the relative merits of using a few units of large size, or a larger number of smaller units, can be assessed theoretically. Thus in geometrically similar engines, inertia stresses will be the same if linear velocities, conveniently denoted by mean piston speed, are the same. Stresses due to temperature differences will not be the same unless temperatures at corresponding points are the same, and when more complicated problems of fluid flow, heat flow, combustion, etc., are considered, theory does not give the complete answer. To provide experimental evidence on such problems, the Massachusetts Institute of Technology (M.I.T.) has constructed three geometrically similar engines of 2½, 4 and 6in. bore, respectively.

Tests under similar external conditions have shown, as expected, the relation of volumetric efficiency to piston speed to be the same in all three engines over a range 480 to 2,040ft. per min. Again confirming theory, cylinder head temperature increases with size, and increases at an increasing rate with piston speed as size increases, resulting in increased thermal stresses. Within the range of size studied, differences in indicated thermal efficiency are smaller than the uncertainties of measurement, and indicator diagrams are identical at the same piston speed. Considering typical commercial engines, spark-ignition engines show a marked increase in knock-limited i.m.e.p. with decreasing bore, allowing increased compression ratio. On the other hand, Diesel engines become less affected by ignition delay as cylinder size increases. The M.I.T. engines show a marked increase in friction m.e.p. as size decreases.

Engines of different size for the same service tend to be rated at the same b.m.e.p. and piston speed, and the output of similar engines at the same piston speed is nearly proportional to piston area, while weight is proportional to bore^3, so that weight/unit output increases with linear dimensions, this tendency being confirmed in practice, in spite of wide design differences. "Life" as limited by wear, increases as cylinder bore increases. Graphs showing the tendencies of specific weight, b.m.e.p. and piston speed with bore size, are given for typical American engines over the range of bore size from 3 to 29in., and appendices give the theory of the dimensional analysis. [M.I.R.A. Abstract No. 5133.]

METALASTIK BUSHES

A GREATLY enlarged edition of the Metalastik bush catalogue has recently been issued by Metalastik Ltd., Evington Valley Road, Leicester. In particular, the catalogue deals with Ultra-Duty bushes, bonded cone bushes for independent front suspension, shackle pins for buses and heavy vehicles, Metacone bushes for engine suspension and cab and body mountings and cone bushes for the independent front suspension of buses and commercial vehicles. These are an entirely new development.

There is much useful information in this catalogue, not only in the form of tabulated data for the various types of bushes, but also in the matter of half-tone and line illustrations and arrangement drawings of various applications. There is also a section dealing with the Metalastik toggle link suspension as fitted experimentally to a B.M.M.O. bus. (1940)

DISC BRAKES

A Résumé of Some Interesting American Design and Development Work

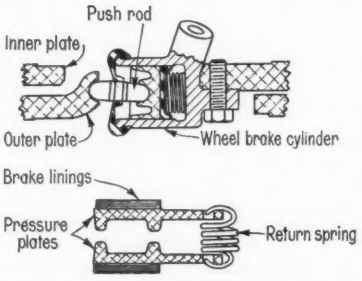
A CONSIDERABLE amount of experimental and development work has been taking place recently in America in connection with disc brakes, both for aircraft and for road vehicles. A very successful application has also been made to railroad equipment. A paper presented to the American Society of Automotive Engineers in March, 1950, read by W. R. Rodger, of the Chrysler Corporation gives an excellent picture of the present position and the following notes are based on the information given in his paper.

The outstanding advantage offered by all disc brakes is the practical elimination of thermal expansion as a factor leading to "loss of pedal". In many designs there is also no appreciable distortion of the parts under load, which again gives reserve of pedal travel, or permits higher mechanical advantage in the actuation mechanism. There is also an increase in brake swept surface by duplication of faces, and this, if a good air flow can be provided, opens up possibilities of greatly improved heat dissipation.

The disc brake therefore can provide greater braking power and will stand up to severe conditions with less rise in temperature of the surfaces and a minimum of distortion, however caused. More self-servo action can in general be provided without erratic action and temperature "fade". Much has been done to develop lining materials that evidence little change in frictional coefficient with rise in temperature. Change still occurs however, and is one of the principal factors setting a limit to self-servo action if consistent operation is to be secured.

It is perhaps not generally known that one important concern in this country, on a vehicle having a powerful mechanical servo, actually uses drum brakes in which all the shoes are trailing in order to obtain absolutely smooth braking, free from shudder and only affected to the minimum by change of friction coefficient.

The disc brake developed by the Goodyear Co., originally for aircraft landing gear, consists basically of a cast-iron disc gripped between one or more pairs of small pads operated direct



Wheel brake cylinder (above) and return spring (below).

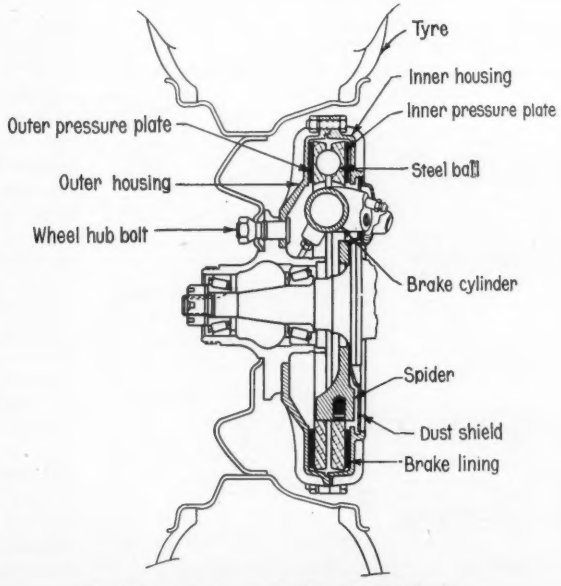
by hydraulic pistons. The larger part of the brake disc is freely exposed to the air and gets the benefit of direct cooling at the point of highest temperature whereas a normal enclosed drum brake can only dissipate heat to the air after it has passed through the metal of the drum. This sets up a temperature gradient which becomes extremely steep when braking hard at high speeds.

It is perhaps not realised that one front wheel drum of a normal car, braked hard from say 70 m.p.h., has to absorb energy at the rate of about 60 h.p. for several seconds. This corresponds to some 40 B.T.Us. per second. If the actual braking surface could be scrubbed with air assumed to undergo a rise in temperature of 100 deg. Fahrenheit in the process, no less than 20 cubic feet per second would be required. Quite apart from the probable damage to the surfaces by dust and mud, the enclosure of the front wheels in the sheet metalwork demanded by modern styling makes the scooping up of such a quantity of air quite impossible.

While the Goodyear car disc brake, which incidentally has only one pair of brake-lining pads, is quite incapable of dissipating the heat of a violent application *as generated*, it was found to get rid of most of the heat of one application before the car had regained speed for a second one. Temperature "fade" after repeated stopping was therefore notably low.

A remarkable feature of the experimental Goodyear brake tested was the small lining area, totalling 9.8 square inches for one front wheel brake, as compared with 50.4 square inches for a conventional 12in. by 2in. drum brake. A very dense lining with a low rate of wear was used. Operation was direct, there was no servo action, but a very high mechanical ratio combined with automatic clearance adjustment.

The brake was tried in two forms in one of which the C-clamp carrying the operating cylinder and brake pads was outside the disc, which was rigidly bolted to the hub while the C-clamp could float laterally. In the other layout, the brake pad assembly came within the disc, which took the form of a ring with external dogs picked up by a spider pressing bolted to the hub, the ring floating laterally and the brake mechanism being fixed.



General arrangement of Chrysler disc brake.

In the Budd railroad disc brake each axle of the vehicle carries two discs, each of which is gripped by two pads operated by levers from air cylinders. The discs are hollow and have Sirrocco-type fan blades cast integral and acting as spacers between the two rubbing rings. A violent radial air flow is thus set up, which, with the considerable cooling surface given by the fan blades themselves, keeps the average temperature reasonably low. This application, which has, it is understood, proved remarkably successful, comes of course, in a different category from normal private car braking. Rates of retardation are probably lower than on a car while the duration of application is greater.

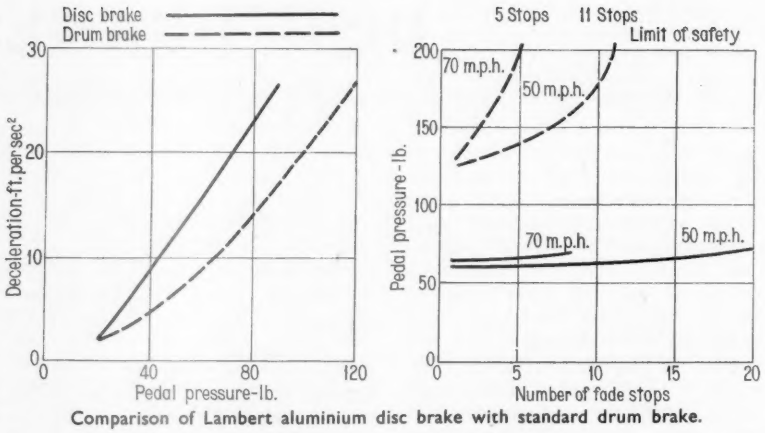
It is reasonable to suppose therefore that the arrange-

ment can dissipate most of the heat as it is generated and that the temperature gradient through the brake discs is low. Maximum temperatures can therefore be kept low, this being much assisted by the large size of the brake discs, which are nearly as big as the wheels on the axle.

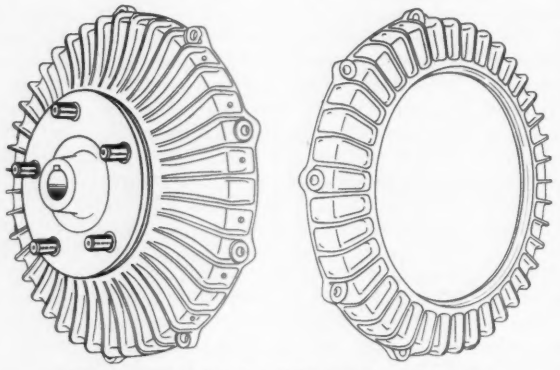
Most of Mr. Rodger's paper is concerned with a description of two similar private car disc brakes, one of which, the Lambert disc brake, appears to be still in the development stage. The other is now in production on Chrysler cars. Basically they consist of two half castings that house opposed annular braking surfaces, spaced about 1½in. apart and engaged by pressure plates. These are faced with bonded friction material and are forced apart by steel balls rolling up inclined planes machined on the inner faces.

One pressure plate is slidably attached to a spider fastened to the axle while the other is rotated in a forward direction by two tangentially-operating hydraulic cylinders. A considerable amount of self-energisation is available, depending on the slope of the inclined planes. In the case of the rear brakes the cylinders turn the two pressure plates in opposite directions, one plate becoming the anchored member in forward rotation while the other takes up the duty in reverse, self-energisation being thus provided in both directions.

The Lambert brake and some of the experimental Chrysler brakes had aluminium alloy housings with cast-iron brake rings secured by setscrews or rivets. Although excellent results were obtained on test, difficulty was apparently experienced in maintaining flatness and intimate contact between the iron rings and the housings. The outer housing is in each case secured to the wheel hub while the inner one extends inwards only as far as the brake surface requires, a small dust-shield completing the enclosure.



Comparison of Lambert aluminium disc brake with standard drum brake.



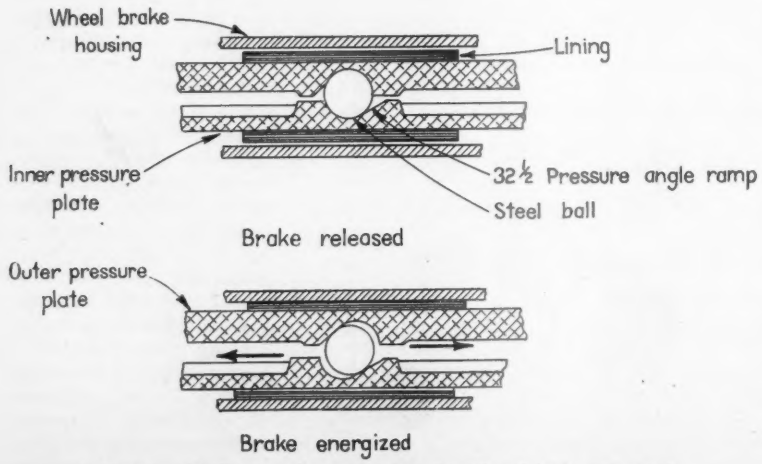
Chrysler outer housing (left) and inner housing.

With the object, no doubt, of reducing to a minimum the moment of the forces tending to "uncurl" the inner housing, the two halves of the Lambert assembly are connected by setscrews, twenty in number, buried in the cylindrical walls. Forty stiffening ribs serve also to increase the cooling surface by about 100 per cent. On the production Chrysler brake, the housings are cast integrally with the braking surfaces in iron. There are forty narrow ribs, giving about 50 per cent. increase of surface, and the housing halves are held together by ten bolts on a rather larger pitch circle.

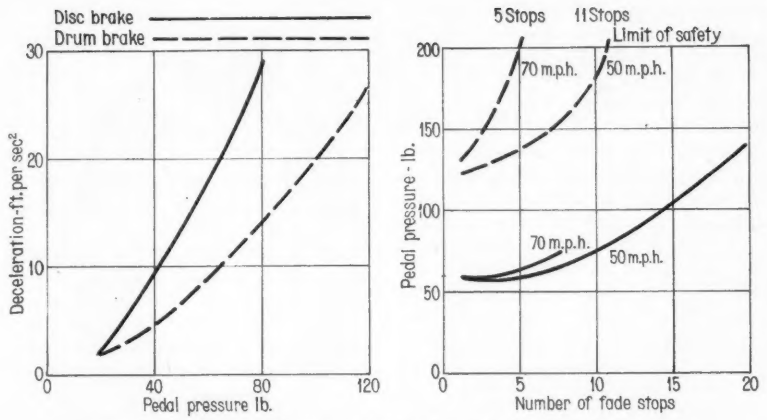
Compared with a standard 12in. by 2in. drum brake, both disc brakes have 12in. external pressure plates of 9in. internal diameter. The housing swept-surface is 99 sq. in., compared with 75.3 sq. in. for the inner surface of the drum. The segmental linings bonded to the pressure-plates give a total of 66 square inches of surface, compared with 50.4 sq. in. for the two 120 deg. linings of the drum brake. The disc brake thus has 31 per cent. more swept surface and 30 per cent. more lining surface than equivalent drum brake.

From an inspection of the illustrations it would appear that the external cooling surface of the Lambert aluminium-housing brake is about 2½ times that of a drum brake, while the Chrysler, with its narrower iron ribs, has about 1¾ times as much cooling surface.

One of the chief improvements sought in developing the Chrysler disc brake was a reduction in temperature "fade". Driving conditions on modern well-engineered mountain roads permit high speeds for short distances coupled with violent braking on the bends. Again, much driving on normal arterial roads consists of acceleration in overtaking followed by violent braking to "pull in" behind the next vehicle, the process often being several times repeated. The increasing use of torque-converter transmissions which



Operation of ball ramp showing brake released and brake energised.



Comparison of Chrysler cast iron disc brake with standard drum brake.

provide very little engine braking is another factor that is throwing more work on the brakes.

Comparative tests were therefore mainly concerned with the increase in pedal pressure resulting from temperature rise and reduced coefficient. They took the form of successive "fade" stops from 70 m.p.h. at one-minute intervals and stops from 50 m.p.h. at 0.2 miles interval. In all cases the pedal pressure required to maintain a retardation of 18 ft/sec. was plotted. The first pair of curves show the effectiveness and the "fade" respectively of a plain 12in. cast-iron drum brake with two wheel cylinders and of the Lambert brake with aluminium housing. The second pair compares the drum brake with the production Chrysler cast-iron disc brake.

It may be seen that the pedal pressures at the beginning of the fade curves for the drum brake do not agree with those given on the curve of effectiveness. Presumably this is because the pedal pressure reading was that at the end of each stop and a considerable amount of fade had taken place. The effectiveness curve is probably taken as an instantaneous reading from all cold.

The much greater effectiveness of the disc brakes is mainly due to the higher actuation ratio made possible by the absence of thermal expansion and deflection under load. Some of the "duo-servo" drum brakes common on American cars give an effectiveness equal to that of the disc brake. Since, however, the self-servo action is much greater the temperature fade is worse. It is clear from the curves that the aluminium Lambert brake has completely solved the "fade" problem, the rise in pedal pressure after eight stops from 70 m.p.h. being almost imperceptible. By comparison the Chrysler brake shows about 35 per cent. increase of pedal pressure, while for 20 stops at 50 m.p.h. the increase is over 100 per cent. compared with about 20 per cent. for the Lambert.

Whether the improvement is actually worth much in normal car practice is another matter. It would seem that conditions of quite exceptional severity would be necessary to show the benefit of the admittedly better cooling of the aluminium brake.

In racing or competition work such a brake might be invaluable. The weight of the car in all the tests was 4,300 lbs. The wheel cylinders on the Chrysler brake were 1¼in. for the front and 1in. for the rear, giving a brake distribution of 61 per cent. front and 39 per cent. rear. The master cylinder was 1in. in diameter and pedal ratio 8.2 to 1.

The output versus input relationships of the brake are of interest. Reference is made to the accompanying ball and ramp diagram and force polygon. Relating forces show that for a 35 deg. ramp (experimental brake) and a friction coefficient of 0.35, the brake frictional force is twice cylinder force. The 32° — 30′ ramp in the production brake provides a brake force which is 2.44 times cylinder force.

Brake designers use what are called an "actuation constant" and an "actuation factor" to measure the degree of self-energizing in a brake. The actuation constant is a mathematical term combining all of the physical dimensions of the brake, exclusive of friction coefficient, which affect self-energizing. The actuation factor is the product of the actuation constant and the friction coefficient; it expresses the portion of total brake input forces which is supplied by the self-energizing feature.

The following equations apply:

For a given brake output,

let P' = brake application force without self-energizing

and P = brake application force with self-energizing

Then, by definition, the actuation factor

$$A = \frac{P' - P}{P'}$$

For the non-self-energizing condition P' becomes the same as $P -/- F$ in the force polygon or

$$P' = \frac{F \tan \phi}{2\mu}$$

and $$P = \frac{F}{2} \frac{(\tan \phi - 1)}{\mu}$$

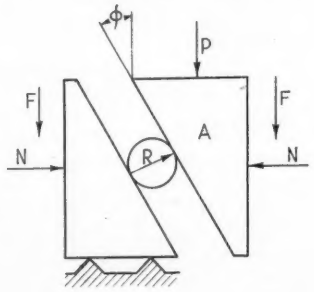
Solving the original equation,

$$A = \frac{\mu}{\tan \phi}$$

and the actuation constant

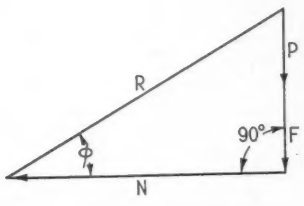
$$Q = \frac{1}{\tan \phi}$$

Hence for a ramp angle of 35 deg.



P = Wheel cylinder force
F = Frictional force on one disc
N = Normal force
R = Resultant force
μ = Coefficient of friction
ϕ = Ramp angle
F′ = Total frictional force on brake

Note: Forces are considered acting at mean lining radius.



From frictional relationship

$F = \mu N$, or $N = \frac{F}{\mu}$

From force polygon:

$$\text{Tan } \phi = \frac{P+F}{N}$$

or $$N \text{ Tan } \phi = P+F$$

$$\frac{F}{\mu} \text{ Tan } \phi = P+F$$

$$F\left(\frac{\text{Tan } \phi}{\mu} - 1\right) = P$$

But $F' = 2F$ (two discs)

Therefore $$F' = \left(\frac{2}{\frac{\text{Tan } \phi}{\mu} - 1}\right) P$$

Ball and ramp diagram and force polygon.

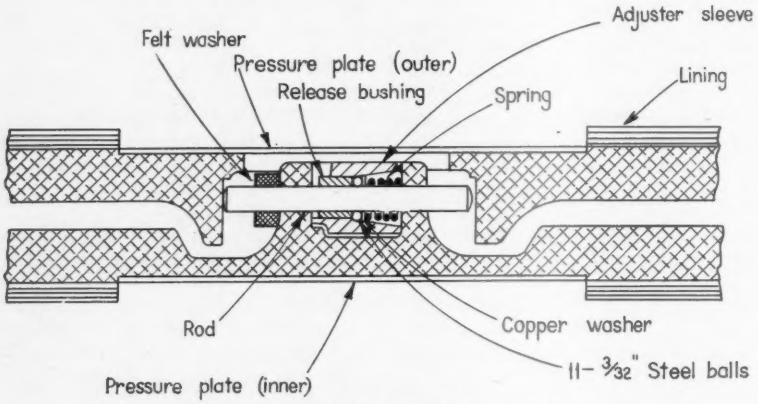
the actuation constant Q = 1.43, and with a coefficient of friction of 0.35, the actuation factor A is 0.50. For 32° — 30′ Q is 1.58 and A is 0.55. In the latter case, the brake is said to be 55 per cent. self-energizing.

Because of the relatively low thermal and mechanical distortions in the disc brake housing, a simple self-adjuster can be used without any risk of "over-adjustment". This takes the form of a pin, sliding in a bracket attached to one pressure plate and having a pre-determined end-clearance between two lugs integral with the other. The pin is held in position by a row of steel balls wedging between it and the conical bore of a bushing between the jaws of the bracket. It is kept under pressure by a washer and an axial spring. This allows movement of the pin by very small increments to take up wear of the lining. Release of the ball grip for assembly is effected by forcing back the bushing with a tommy-bar.

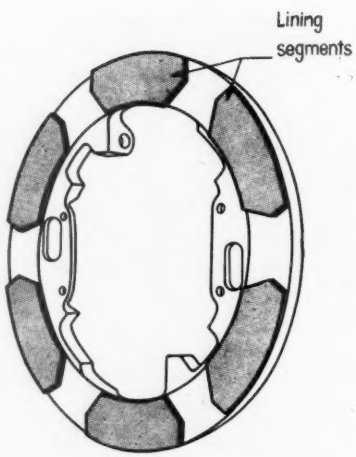
It is interesting to note that the original adjuster had a sliding pin held by a "cocked" washer and off-centre spring. It was found, however, that the inertia of the washer was sufficient for it to move on a rough road and release the pin. This had disastrous results, as the brake acquired enough clearance to bring the pedal to the floor.

Much development work was done on the brake lining. The first brakes had continuous rings of material and suffered badly from scoring owing to the absence of an escape path for abraded material. So severe was the action that at times pieces of the lining dust would "ball up" and completely tear the lining off in a few stops. A solution was found in the use of six segments 4in. long with considerable gaps between them. The lining is bonded to the pressure plates and six segments can be blanked from a sheet 4in. wide by about 9in. long with very little scrap. The bonding cement is applied to the sheets before blanking.

Practical experience in manufacture emphasised the vital importance of accurate machining of the housings as regards parallelism of the friction surfaces. Great accuracy was also found to be necessary in the machining of the ramps for the balls. In production these are machined at one setting by six spherical end-mills having a diameter .008in. larger than the balls. The cutters are fed towards the plate on a helical path.



Self-adjusting mechanism with brake applied.



Chrysler outer pressure plate.

As previously mentioned, both pressure plates in each rear brake have angular freedom until brought up by one or other sets of stops according to direction of rotation. Considerable difficulty was experienced in eliminating the "clocking" noise as the brakes reverse direction. A solution was found in the use of spring plungers cushioned by Belleville washers giving a constant load for the major part of the movement with a sudden increase just before "bottoming".

In the case of the front brakes the inner pressure plate is loaded against its stops by a spring to check noise. Spring-loaded plungers are located in the two lower spider arms to prevent the pressure plates from rattling on rough roads.

The author of the paper states that the disc brake in its present form, is considerably heavier and more expensive than the drum type. It will not therefore be applied to cars in the lower price ranges. Development work is however now in progress with designs offering a promise of simpler construction and lower cost.

Leaded Alloy Steels

IN the production of many components, the machining costs represent a high proportion of the total. For many such components the use of an alloy steel containing lead may well lead to a reduction in final cost, despite the increased initial cost of the material. That this is realized, particularly in the automobile industry, is shown by the increasing use that is now made of these steels.

For the benefit of engineers and metallurgists who may be considering the employment of alloy steels containing lead, a booklet has been published by Samuel Fox and Co. Ltd. that gives in concise form information that is often required. Two tables of particular interest are included. The first gives comparative mechanical test results for some of the more widely used lead-free and lead-treated alloy steels. The second table gives comparative production results on a wide range of automobile components. Copies of the booklet may be obtained on application to Samuel Fox and Co. Ltd., Stocksbridge Works, near Sheffield or to the Publicity Department, The United Steel Companies Ltd., 17 Westbourne Road, Sheffield 10 (P.O. Box 64).

Standards for Magnetos

A NEW British Standard B.S.1644 : 1950 concerns the dimensions of unscreened camshaft speed magnetos. It deals only with magnetos combined with distributor head, having shank and spigot mountings, and the dimensions included are those necessary to ensure interchangeability. Copies can be obtained from the British Standards Institution, 24 Victoria Street, London, S.W.1, price 2s. 6d. post free.

The Rockwell Machine Tool Co. Ltd.

THIS Company has now been appointed Sole Agent in the United Kingdom for the "Jung" internal grinding machines manufactured in Berlin. Many users of "Jung" surface internal grinding machines will remember from pre-war days the qualities of these machines. The main Jung Works in Berlin survived the war, and the Company has now been re-formed in the Western Sector of Berlin. The Management and skilled staff are mainly the original body of workers, thus ensuring a continued standard of design and workmanship. At the present time only one size of internal grinding machine is available for early delivery, but larger sizes will follow successively.

HONING EQUIPMENT

The Development of More Powerful Tools for Faster Stock Removal

HITHERTO the honing of internal bores has in this country been regarded mainly as a finishing operation for the production of accurate and highly finished surfaces on bores that have previously been machined to fairly close limits. There is now however, an increasing tendency to effect an overall economy by eliminating the final boring operation and by using much greater power at the honing operation to remove as much as 0.020 to 0.030in. of stock from a relatively rough bore.

To meet this demand, George H. Alexander Machinery Ltd., 82-84, Coleshill Street, Birmingham, have recently introduced more powerful A-B hones and drivers, some of which are capable of transmitting from 20 to 60 h.p. according to speed. These were designed primarily for export to the United States of America where the tendency to use hones for heavy stock removal originated, but two types are now available for users in this country. It should be mentioned, however, that correspondingly powerful honing machines are necessary if full advantage is to be taken of the new equipment.

Until recently European honing machines have rarely been fitted with motors of greater power than 7½ to 10 h.p. for the general run of cylinder and light diesel liner work up to about 10in. diameter. For larger diameters up to 36in. many organisations use either converted lathes or boring machines for driving the hones.

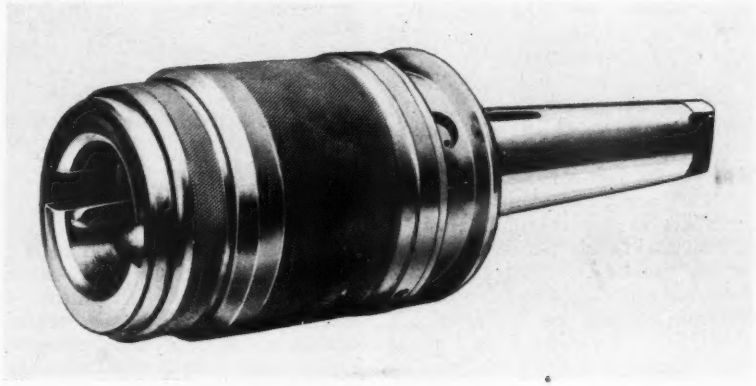


Fig. 1. A-B drivehead HD 10-80 for heavy duty hones up to 8½in. diameter.

The new A-B model HD 10-80 drivehead shown in Fig. 1 is suitable for driving and controlling hones of the type shown in Fig. 2 up to 8½in. diameter. These two units are coupled in the usual way by a universal, with universal joints at each end to compensate for mis-alignment. The universal is made to suit the type of hone and the diameter and length of the work. In principle, the new tools follow closely on the lines of previous A-B designs. The drivehead is provided with planetary gearing to expand the hone while running and to collapse it when the honing is completed. An adjustable sizing stop and a graduated ring are fitted to prevent honing oversize and to give duplication of size without undue gauging on successive bores once the correct allowance for abrasive wear has been determined.

An important point of difference is that the coupling universal is detached from the drivehead instead of from the hone and a quick-action automatic lock is arranged to secure the ball-end of the universal. This locking device is comprised of sliding keys projecting into the driving bayonet slots. The keys are controlled by the smaller knurled ring shown in Fig. 1. When this is moved axially towards the shank, the keys are withdrawn and the universal can be removed. In coupling up, the action of entering the universal raises the keys, and the ring, until the ball-end is turned into safe driving engagement. The keys then drop into the locked position.

Expansion of the hone is effected in the usual way by means of box spanners and a square shaft extending through the universal to a centre spindle in the hone, on which two opposed cones are drawn together by right and left hand threads. The complete cone assembly is centralised by a simple patented device that permits approximately 0.010in. axial float, under strong spring control to bring the cones back to the central position from displacement in either direction. This end float of the cones is essential to allow the abrasive stones, which are expanded by thrust pins bearing on the cones, to take up a slightly conical formation in a tapered bore. This formation equalises the cutting stress and wear throughout the length of the stones. In this way, the fastest possible correction of errors in parallelism is effected. It ensures that a heavy cut is taken at the small end of the taper and little or nothing at the large end, whilst the stones automatically take up a more and more parallel position on successive strokes until both the hone and the bore are truly parallel.

This action has been explained in some detail because it is a common error to assume that the stones should be expanded parallel in order to produce a parallel bore. However, a little reflection will show that parallel stones in a bore with errors of taper, will cut on only one end. At best, this will result in uneven wear on the stone and defeat any efforts to produce a parallel bore. If there are serious errors in the bore, the stones will chatter and break.

In passing, it may be noted that when honing stones chip or break owing to careless operation, it is always the trailing edge of the stone

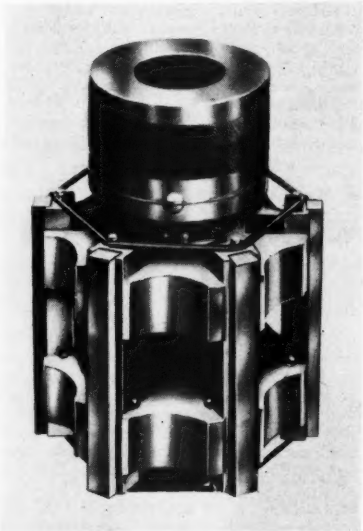


Fig. 2. A-B heavy duty hone for bores up to 8in. diameter.

that chips away. With this in view, all A-B hones are made with the stones slightly offset in advance of the centre. This concentrates the pressure of new stones on the front edge, which is well supported, and protects the rear or trailing edge. As a result, heavier and faster cuts can be taken with safety.

New types in the larger range of A-B honing equipment are shown in Figs. 3 and 4, which show respectively the H.D. 70-150 drivehead and the H70-150 hone. These have a diameter capacity range of from 7in. to 15½in. They are suitable for a variety of diesel, pump and compressor liners or cylinders. Provided sufficient power is available to allow full use to be made of this equipment, very high rates of stock removal can be maintained. For example, it is now common practice in the U.S.A. to remove from 1 to 2½ cubic inches of metal per minute by honing according to the nature and hardness of the material. This gives an average of over 0.001in. per minute from the bore of an average diesel liner 10in. diameter and 42in. long. An approximate honing time can be calculated for any size of liner by comparison with the above figures, bearing in mind that increase or decrease in diameter and length will affect the honing time in inverse ratio.

In Fig. 3 the HD 70-150 drivehead and expansion unit is shown with coupling universal and the female driving coupling on the end of the hone. An interesting new feature of the design is the method of taking the axial thrust or tension in the ball-couplings without impairing the universal action and without complicated latches or split collars etc. to allow couplings to be instantly engaged or disengaged.

For this purpose the external and internal spherical members have an interrupted surface, so that in the disengaged position, the spherical projections on the ball-end come opposite the slots in the housing. The two parts can thus be coupled up, and when the driving pins are turned into the bayonet slots through an angle of 45 deg., the spherical portions are brought into full engagement. A shroud ring supports the housing against any tendency to open at the bayonet slots as a result of the heavy axial load when the coupling is in tension.

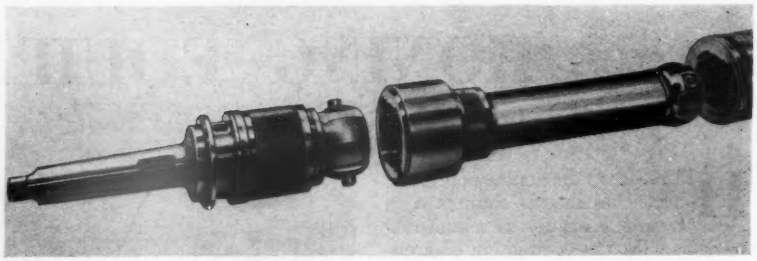


Fig. 3. A-B drivehead and expansion unit for hones up to 15½in. diameter.

A similar coupling is provided at the other end of the universal, the two end members of which are made with accurate spigots between which a tubular member may be pegged and brazed in position. By this means, the universal may readily be made of any desired length to suit the work. The usual box-spanners and square keys, connected by a smaller diameter tube through the universal, couple the adjusting mechanism of the drivehead to the cone-screw in the hone.

Both A-B driveheads have provision for the attachment of a brake or hand-wheel (supplied as extras) for facility in operating the control sleeve while running. These attachments also serve to give extra power when applying the heavy cutting pressures required for honing hard materials or chromium-plated liners.

With the type of hone shown in Fig. 4, a wide range of diameters may be handled with one basic hone, by fitting flanges of the required size. They are jig-bored so as to locate on hardened and tempered high-tensile fitted bolts, ground to close limits, and on spigot diameters on the body. Provision is made to ensure that the top and bottom flanges can be assembled only in the correct order and position. Eight sets of flanges and expansion or thrust pins cover all diameters from 7in. to 15½in. in 1in. steps. Intermediate sizes are obtained by the use of interchangeable standard and oversize stone holders, which are hardened and ground to fit the slots in the flanges.

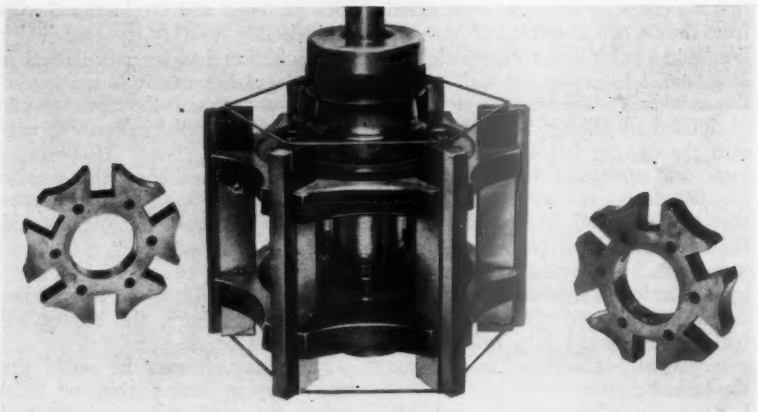


Fig. 4. A-B hone with interchangeable flanges for all diameters from 7 to 15½in.

Aluminium in Road Transport

PUBLISHED by the Northern Aluminium Company Ltd., Banbury, *Aluminium in Road Transport* is an 84 page publication dealing very thoroughly with this subject. Following a discussion on the advantages of aluminium, are sections on goods vehicles, public service vehicles and private cars, showing not only typical examples of finished products, but also many constructional details. There follow sections on power units and transmissions, internal and external fittings, designing in aluminium and building in aluminium. These last two sections are concerned with mechanical properties, available forms, principles of design, machining, forming and bending, joining, and priming and painting. The final section deals with repair and maintenance and there are useful appendices giving information of value to the designer. This booklet is extremely well produced and is lavishly illustrated. (1934)

Klinger "Sleeve-Packed" Cocks

RICHARD Klinger Ltd., Klingerit Works, Sidcup, Kent, have recently placed on the market the AB10 cock to replace small, taper plug cocks for pressure gauges, air reliefs, drains, instrument connections, compressed air, gas and hydraulic controls and similar equipment. This is a simplified design with the minimum of component parts. The main body is machined from a hot brass stamping and is polished all over. Tested to 1,000 lb. per sq. in., it is available in six different patterns, with a bore of $\frac{1}{32}$in., having $\frac{1}{4}$ or $\frac{3}{8}$in. B.S.P. external or $\frac{3}{8}$in. or $\frac{7}{16}$in. Whitworth internal screwed connections, provided with or without union ends. (1935)

WELDING*

The Modern Application of the Metallic Arc Process

IN welding plant for the metallic arc process the ideal required is a constant current with varying voltage, which must be quite high for striking the arc and then show a drop to a figure determined by arc length. This in turn is dependent upon the flux coating of the electrodes. For operational safety an upper limit has been placed on the open circuit voltage of both A.C. and D.C. welding sets. It is 100 volts D.C. or R.M.S. value A.C. In some circumstances the open circuit voltage is reduced to as low a figure as 60 volts. In practice there must be a sloping characteristic so that the current may be controlled. As the voltage falls the current increases but the voltage being insufficient to support the arc the current returns to normal. A similar process controls it in other directions.

Despite the efforts of electrode manufacturers, they have been unable up to the present to produce an electrode suitable for welding some of the non-ferrous alloys using A.C. supplies. Recently engineers have produced a unit that generates a high frequency current, about 0.5 megacycles, of high voltage and negligible power which is superimposed on the main current form. This obviates the restriking troubles and even starting difficulties on very light gauge wires, 20 g., so that with this equipment any of the metals previously welded only by D.C. are within the capacity of A.C. equipment.

Another development is electrodes with deep penetrating qualities which are obtained by using a high arc voltage. This is effected by making the coating burn-off rate slower, thereby increasing the arc length and the depth of the flux cone. Another new electrode is one that can be used in deep grooves because instead of producing sharp undercutting it produces a weld that is deposited equally up the sides. These electrodes use a flux containing a large percentage of iron oxide. Some makes are able to satisfy the deep penetrating requirements, but iron oxide cannot always be relied on to impart these properties. This point is important for two reasons. First, those electrodes employing cellulose for the deep penetrating properties are adversely affected by overheating. That is, they lose their deep penetrating properties if the welder tries to use a higher than recommended current in an attempt to obtain greater power and hence greater penetration. The iron oxide type operates in the reverse way. Secondly, the cellulose type produces sharp undercut and notch effects, which unless removed considerably reduce the notch resistance of the weld. Iron oxide rods do not.

Automatic processes

Automatic processes may be divided into those using continuous electrodes and others. By far the most important and successful are the continuous type, which again are divided into two major processes, (a) submerged arc or Union melt process, and (b) covered electrode type as developed by Fusarc and Metropolitan Vickers. The others suffer from a difficulty in reloading individual electrodes without breaking the arc. To overcome this, the machines are necessarily more complicated, with a greater number of parts that may go wrong in service.

One other form of arc welding used for mild steel that should be mentioned is the process known as stud welding. This requires a different current voltage curve, but the metallurgical features are the same as in metallic arc welding. The main difference between the processes advocated by different manufacturers is the method of applying the flux. In this process, the function of the equipment used is to regulate the heat by controlling the amperage and time which are set for the varying operating conditions. An arc is drawn between the stud and the parent metal after which the stud is plunged into the pool of molten metal. A porcelain surround prevents the molten metal from splashing out. A recent American development of this process is a method of arc spot welding on thin plates in which an arc is drawn in a similar gun between the parent metal and a tungsten electrode surrounded by an atmosphere of inert gas. Although the application of this process is limited, it is very suitable for quickly spot welding assemblies that are too big or awkward to be handled in an ordinary spot welding machine.

Metallurgical considerations

An ideal weld is one that has the same physical properties as the parent material, but in practice this is very difficult to obtain owing to the metallurgical changes in the structure of the parent and weld materials in the heat-affected zone. In order that undesirable products of the transformation may be avoided, careful consideration should be given to the welding procedure and post-welding heat treatment for each particular material composition. To make a weld, heat is used to melt the parent materials locally and fuse them together, often with the addition of a filler material. This local heating produces a very severe gradient which causes changes in the heated zone. These are, structural transformation, crystal size and thermal expansion and contraction.

Structural changes take place when iron or plain carbon steel is heated and cooled, but differing physical properties are characteristic of certain structural forms. From the welding engineers' point of view the austenite transformation is the most important, since it determines the physical properties at ordinary temperatures. This is decided by the rate of cooling. The trend is for the time required for the reaction and the hardness of the products to decrease as the carbon content is reduced. The significance of this is important since the ductility usually increases as the hardness is decreased.

The crystal size depends upon, (a) size of austenite crystals, (b) amount of cold work and (c) size and amount of inclusions. When a form of iron is heated to a temperature approaching its lower transformation point, crystal growth begins. The rate of crystal growth increases until the lower transformation point is reached when recrystallization refines the grain size. The large austenite crystals retard nucleation and complete transformation during the subsequent cooling cycle.

A steel that is cold worked has elongated crystals produced by twinns and slips along cleavage planes. These occur during plastic deformation. Such crystals have large ragged boundaries that favour re-crystallization at a temperature below the lower critical AcI. Hot peening makes use of this property to produce a fine-grained weld, but it requires very strict control, otherwise cracking will occur if the weld tends to be blue brittle. Inclusions retard grain growth by obstructing atomic movement in solid

* From a paper presented to The Institution of Production Engineers by G. Cubitt-Smith.

solution, but they are undesirable since they often contain objectionable alloying elements. They also promote nucleation but increase transformation time.

Induced stresses

The volumetric changes produced by heating followed by cooling can cause two types of stresses, (a) shrinkage and (b) residual. One of the physical properties of metals is to expand on being heated approximately linearly until they reach a transformation temperature where a sudden change in volume occurs. Alpha iron occupies a greater volume than gamma from which it transforms in cooling, but martensite has the lowest specific gravity and greatest volume. As the martensite transformation takes place at a relatively low temperature, 150 deg. C., the metal is fairly hard. The reaction produces high compression stresses and the expansion may result in cracked crystals. This weakens the metal and may cause failure as contraction stresses increase again, or under sudden overloads in service. After the reaction at 150deg.C. normal contraction continues until room temperature is reached.

These crystal cracks are not visible except under a microscope. Therefore special precautions must be taken in welding steels liable to form martensite in the heat-affected zone. The remedy is to pre-heat to 200deg.C. to prevent the formation of martensite until the welding is completed. The welded assembly can then be cooled slowly through the martensite transformation range or, preferably, annealed without cooling to change the iron form. Annealing will not heal crystal cracks once they have formed.

Linear contraction stresses the welds which may cause distortion in the welded components if proper precautions have not been taken. Otherwise the weld metal should flow plastically, and when the yield stress is reached it will not crack unless the metal has blue brittle tendencies. Up to a short while ago engineers believed that residual stresses added to load, and this phenomenon has been used to account for failures in the past. However, this theory could not be proved practically, and it is now known to be false.

These stresses are not detrimental to the load carrying capacity of the weldment, nor do they seriously impair the ductility of the material. They will, however, affect slight dimensional changes when the weldment is machined. Therefore it may be more economical to stress relief the component rather than resort to a preliminary machining, then re-setting and machining again two or three times with a reduced depth of cut each time. As a general rule it is advisable to stress relieve all machined tool components and other components where slight movement of machined faces may be detrimental.

Effect of alloys

Carbon. This is easily the most important alloying element in steel. It should be kept below 0.25 per cent. for welding.

Sulphur. This forms an iron sulphide that solidifies at 925 deg. C., promoting hot cracks by extension of the mushy state. 0.05 per cent. S can be regarded as a maximum.

Manganese. Since manganese combines with sulphur to form a harmless compound, manganese sulphide, it is in some respects a remedy for excess sulphur. If the sulphur effects are to be completely eliminated the manganese must be present in quantities three times that of sulphur. Manganese is also slightly more prone to form carbides than iron but is mostly present dissolved in ferrite at low temperatures. It is easily burnt in long puddling periods, which should be avoided in the welding process. Manganese in excess of 1.0 per cent. should not be melted without special precautions being taken to avoid deep hardening.

Phosphorus. This produces banding, which makes material weak in certain planes. For welding the allowable maximum is 0.06 per cent.

Silicon is a powerful deoxidiser and dissolves in ferrite.

Copper does not form carbides and will only dissolve in ferrite up to 0.8 per cent. maximum. This figure should not be exceeded for normal welding purposes.

Aluminium. This is a powerful deoxidiser, hastens transformation of austenite and increases fluidity.

In this country three types of metallic arc welding are in general use. They are: (a) light current (multi-run), 300 amps and below, (b) heavy current including automatic and (c) austenitic welds. The tendency is for heavy current welding to be used where possible. This is a point of great controversy between engineers. From metallurgical considerations a compromise between multi-run and heavy current welding will probably prove to be most satisfactory.

In multi-run welding, the first run is fully annealed by the second, and so on as the number increases. It can be performed in any position, but suffers badly from cracking due to low heat input and chilling of the weld, which probably causes the formation of objectionable martensite. Low penetration prevents the removal of mill scale and rust by flotation into the slag. Instead it enriches the junction with these undesirable elements which may cause cracking. There is also greater distortion due to the accumulative effects of each run. In heavy current welding there is less distortion, good penetration and removal of mill scale and less chilling of the weld. It is also cheaper when manipulation is not necessary. On the other hand, it suffers from lack of annealing when welds consist of a single run and is very limited in welding position, only down hand being really suitable.

Automatic welding is used on very rigid components. Where high weld ductility is necessary it may be obtained in the 18 per cent. Cr., 8 per cent. Ni electrode which is austenitic at all temperatures. Care must be taken to avoid excessive penetration since the weld metal will be diluted by mild steel to a dangerous percentage (9 per cent. Cr, 4 per cent. Ni, an air-hardening steel). This state of affairs will produce martensite in the junction area, but the layer is so thin and distributed that it is of no consequence. These welds are extremely hard owing to carbide precipitation due to the higher carbide forming tendency of chromium. These should not be used in applications that call for later machining of the weld.

To summarise, it is essential to ascertain the metallurgical changes that may result in undesirable forms of steel in the weld zone. They are determined by the number and quantities of alloying elements present and the cooling rate of the weld, dependent upon the mass of surrounding material receiving heat by conduction. Where possible, heavy current welding including deep penetration should be used, and a second run should be made where welds are heavily stressed. When positional welds have to be used the parent metal should be cleaned of all scale and thick sections pre-heated from the opposite side. For rigid components, austenitic welds should be used and with them there should be careful adherence to the instructions of the welding engineers.

The Mond Nickel Company Limited, Sunderland House, Curzon Street, London, W.1, have recently issued *Aluminium Alloy Castings Containing Nickel*, based upon a paper presented by Frank Hudson to the Congress of the American Foundrymen's Society in May, 1950. The subject is dealt with very completely, beginning with casting alloys in use to-day and continuing with notes on casting quality, metallurgical developments and production methods. Much useful data are tabulated and there are, in addition, numerous valuable illustrations.

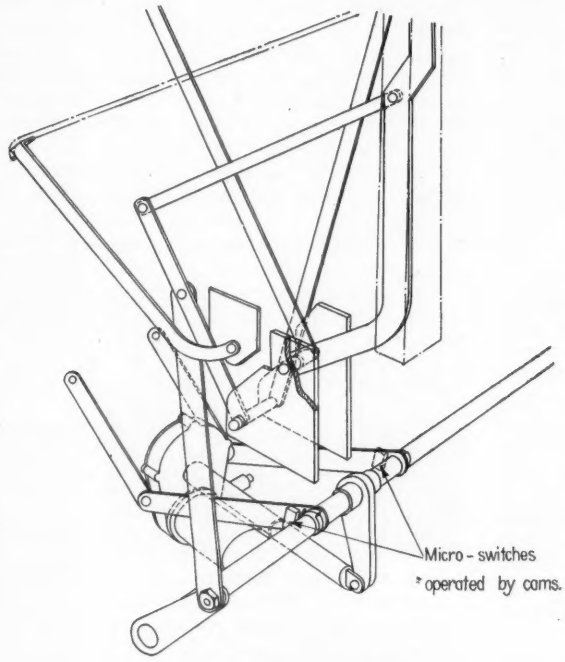
POWER-OPERATED HEAD

Details of a New Electrically Operated Mechanism

POWER operation for the head linkage on full convertible coachwork is now an accepted feature on such types of body, and frequently apparatus of this sort is included in the specification of convertible cars exported to warm countries. It is therefore interesting to consider the features to be found in a mechanism which is purely electrically operated.

An electrically operated mechanism which has interested some coachbuilders in this country is that evolved by P & D Motor Development Co. Ltd., 139, Kensington High Street, London, W.8. This system comprises an actuator which embodies a 12-volt series wound motor providing linear thrust through spur gear reduction. This drives a lead screw engaging with a phosphor bronze nut let into an 8in. ram. The thrust applied in either direction by the actuator is transmitted through the medium of a cross shaft, to cranks at either side of the car, and thence by means of a connecting rod to an orthodox type of head linkage.

The accompanying illustration shows the layout of this equipment. The view is that of a scheme applicable to a car having accommodation beneath the rear seat board for the essential units. It is quite possible however, to provide the movement to the head linkage by suitable connection with the actuator unit mounted in some more convenient position, convenient, that is, for the interior layout of the vehicle. It might for example, be found that insufficient room is available beneath the rear seat board, in which case the luggage boot offers a possible location. A well swept streamlined type of rear wing in which the rearward half is often wasted, would accommodate the actuator, which is of course required only on one side or the other of the car.



Layout of P & D hood mechanism

Control is by means of a press button mounted conveniently to the driver, this remote control operating twin relays, whilst an automatic cut-out is provided by the tripping of micro switches which are actuated by cams located on the cross shaft.

In the installation of this electric hood mechanism, the cross shaft may in some cases present a problem, but the efficiency of the mechanism is not impaired by the cranking of this cross shaft, to give the requisite clearance for the propeller shaft, for example. The side links which couple the head mechanism to the cranks at the ends of the shaft, can in practically all cases be concealed in the quarter trimming at either side of the rear seat. Although no such device is shown, it would be advisable to make provision for the easy detachment of these operating links to ensure that in the event of failure, it would still be possible to operate the head linkage manually. The upper ends of the links should have this detachable device, and the lower quarter trimming should also be readily removable to facilitate this operation.

The thrust applied by the actuator is in the region of 1,000 lb, and the operating time for raising and lowering is approximately 5 seconds in each direction, although this figure can of course be modified and would probably be improved if a time cycle of 8 to 10 seconds were involved.

Similar equipment on a smaller scale has been developed to supply a double-acting electro mechanical system, which can be applied to a panelled hood cover. This could be raised when the hood is moving either upwards or downwards, covering the top of the hood when the hood is fully folded, giving a complete concealment. Suitable tripping switches would give co-ordinated movement, providing entirely automatic operation of such a hood cover.

THE PERKINS P3(TA) DIESEL ENGINE

A NEW three - cylinder Perkins diesel engine has been announced recently, developing a maximum of 32 b.h.p. at 2,000 r.p.m. and a maximum torque of 88 lb. ft. at 1,200 r.p.m. It is stated to be particularly suitable for use in conjunction with the Ferguson tractor.

Following the already well-known "P" series design, the engine has a bore and stroke of 3½in. by 5in. and a swept volume of 2.36 litres. The cylinder block and crankcase are cast integrally in alloy iron, while the cylinder head is a chromium iron casting in which the valves and tappets are carried directly. The camshaft is mounted high up on the offside of the cylinder block. Both main and big end bearings are of the shell type and are steel backed with copper lead linings. Chrome molybdenum steel stampings are used for the connecting rods. For speeds up to 1,500 r.p.m., cast iron alloy pistons are employed, while for higher speeds aluminium alloy is specified.

Enclosed in a timing case at the front of the engine, a triple roller chain drives the camshaft and C.A.V. fuel pump. The injectors mounted in the throats of the combustion chambers give two sprays in accordance with the "Aeroflow" combustion chamber design which is a Perkins patent. The cooling system embodies a pump.

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

B.R.M.

SIR,—I beg to refer to your remarks on the B.R.M. in the *Automobile Engineer*, volume XL. No. 533, November, 1950, page 349/350. This editorial should first be met clause by clause. (A) Grand Prix racing may be a sterile field so far as the main trend of automobile engineering is concerned, but the fact remains that it is a thriving sport supported by various organisations in a number of different countries. The technical benefits arising from success (and from failure) far from being debatable are essential if further success is to be obtained and failure avoided. (B) It is easy to say that the finance involved would be far better spent in co-operative research, but this could equally be said of the millions spent on football pools by the members of the motor industry; furthermore it might be said that if such finances were devoted to co-operative research they could much better have been spent on agricultural development, in fact there is no end to this argument. In any case if co-operative research wants money as badly as the B.R.M. it stands at least as good a chance of getting it if the industry really feels that it equally deserves it.

(C) The motor and allied industries have provided the bulk of the funds, many components, and much equipment for the production of the B.R.M. and the Government have given special priority for the supply of materials so that there would appear to be some foundation for the widespread impression of co-operative effort and national support.

(D) The editorial inference is that discredit should be centred on certain individuals at Bourne and not be permitted to reflect on British engineering as a whole. What however, are the facts surrounding the failures experienced on the race track to date? Of the occasions where the car has had mechanical failures in public so far, two of the failures were of specialised materials or components not made at Bourne, but by specialised firms who are members of the British motor car industry. One failure was due to the escape of gear box oil through the hole left when a relatively unimportant stud came out of the housing. Blind holes are the answer for such studs, and the editor could usefully educate the whole industry in this respect; Bourne certainly will not forget.

(E) To say that there is nothing about its performance so far, to suggest that in present hands, the ultimate success of the car can with any confidence be anticipated, implies in one paragraph that the car is not fast enough and that those at Bourne are incompetent to bring it to the required pitch of performance. It is quite unreasonable, at any rate to any engineer who has examined in detail this masterpiece of a car to think that those who have conceived and produced it are unfitted to bring it to a state of perfection. Development has been hampered by an insufficiency of engines to be able to afford to write them off on the test bed, and an insufficiency of complete cars to be able to carry out extensive handling and performance tests. (F) The will, the knowledge, the experience and the facilities are all there at Bourne, all that is lacking is enough money. There is no question of graceful withdrawal being impossible or of it being too late to allow the B.R.M. to pass into the limbo of projects best forgotten. Only more funds can prevent this from happening.

(G) There is plenty of determination to fulfil all the hopes expressed, but financial support is indeed sorely needed, and if the editor is sincere in his appreciation of the situation, he should pause to think whether his ensuing remarks are best calculated to assist in the raising of that cash. He has expressed fears of discrediting the motor industry and then proceeds to do what appears to be his best to discredit the real fountain head of the project. His remedy for what he regards as an unhappy reflection on our technical reputation and upon British Industry is to place the project within one of our factories, presumably a car factory. It would be interesting to know whether he has in mind a large production works many of which have been so generous and helpful but who have no experience whatsoever of motor racing, or whether he would go to one of the specialist sports car firms, supporting the project, who may have limited resources but vast enthusiasm and experience of racing. In any case considerable expansion and probably a completely new department would be needed by any firm who took on the development of the B.R.M. and it would inevitably cost very much more than proceeding with the existing and very adequate installations at Bourne.

(H) It is the easiest thing in the world to criticise, deride, and build castles in the air, but one of the hardest must be to raise a quarter of a million pounds and with it to bring the B.R.M. into existence, and it would seem to be the height of folly to cast aside those who have achieved this task, in favour of a motor factory which could have gone in for Grand Prix motor racing at any time had it seen fit to do so.

IVAN M. WALLER, B.A.,
A.M.I.Mech.E.
Technical Manager.
Small & Parkes Limited.

(Following our correspondent's "clause by clause" suggestion, his chief points are lettered for reference.

(A and C) Our correspondent confirms our viewpoint, which is that the project has come to be associated with the industry as a whole, and failure is the more serious on that account.

(B) This contention is fallacious, being based on subjects so remote as to be valueless for purposes of analogy. In any event there is no point in clouding the issue, which is a very simple one. It is to make the best use of the industry's resources for the benefit of the industry's dollar earning products. Football pools, flying saucers, Christmas ghosts, etc. have nothing to do with this.

(D) Final responsibility must always rest with those who are in fact responsible. Failure cannot be passed back to industrial suppliers, who work to drawings and specifications or other requirements laid down by those in charge.

(E and F) Our comments imply precisely what was stated, and follow the facts to date. These comments specifically commend the basic layout while criticising the manner in which the project has, in every other respect, been handled.

(G) The project should presumably go to an organization competent by personnel and equipment to handle it. The requisite financial backing is much more likely to be forthcoming if

supporters have sound reason to believe it will be usefully employed.

(H) It is precisely because the present directorate has *not* "achieved its task" that drastic change is essential. —*Editor.*)

FRONT SUSPENSION

SIR,—Having read your very interesting and comprehensive Motor Show report in the November 30 issue of the *Automobile Engineer*, we note that you consider our front suspension "to be quite unsuited to modern private car conditions, where far more flexible suspension and absence of fierce damper settings is essential."

We would like to point out that the front wheel deflection is 3 in. to full bump and 2½ in. to full rebound, as wide a range as several wishbone arrangements and greater than the vertical pillar arrangements. There can be no doubt but that this range is fully and regularly used.

The deflection of the spring is 4 in. from free to static height and the relation between spring and wheel on the axle beam is 1 to 1.70, which gives, of course, a static wheel deflection of 6.85 in. and a periodicity of 71.5 oscillations per minute.

The shock absorber settings give quite nominal torque figures and the mounting position is such that the minimum necessary for satisfactory damping may be used, ensuring, of course, lowest possible internal pressures. We may add that the front end of our cars is considerably softer than one wishbone suspended model, the weight of which is just over half that of our saloon model. Another wishbone suspended model now employs two 1 in. bore shock absorbers each side.

The larger shock absorbers (1⅜ in. bore) on our 'J2' model have the same torque figures as the 1 in. bore units on the saloon and 'K2' models. The purpose of the large units is to ensure reliability under competition racing conditions.

Although the geometrical layout of our front suspension would appear to give excessive tyre scrub, the tyre life under conditions of very hard driving is 14,000—15,000 miles, not bad by wishbone standards.

ALLARD MOTOR CO. LTD.
R. J. CANHAM,
Director and General Manager.

A MAXIMUM PRESSURE INDICATOR

A Simple Device for Internal Combustion Engine Development

MAXIMUM and compression pressures of internal combustion engines can be measured by a new instrument manufactured by Dobbie McInnes, Ltd., 191-3, Broomloan Road, Glasgow, S.W.1. It is known as the Dobbie McInnes peak pressure indicator, Fig. 1, and is being made under licence from the inventors, Sulzer Brothers, Ltd.

Fig. 2 is a diagram of the working principle. The indicator is fitted with fixed connections or coupling nuts to fit any well-known engine indicator valve. Cyclic engine pressure acts on the outside of a two-ply stainless steel bellows unit A, which is sealed at one end to a push rod B and at the other end to the outer shell. Pressure applied to the bellows raises push rod B against a calibrated spring C. A conical nut D screwed to the top of the push rod by a quick thread is also pushed upwards, but is also wound downwards to its seat again under the action of a pre-loaded clock spring F. Rotating with the nut is a thimble, the rim of which is graduated in English or metric pressure units, so that the nut and thimble indicate the vertical travel of the push rod after the manner of an ordinary micrometer screw gauge.

During the first engine cycle, after opening the indicator valve, the nut does not have time to lock the push rod in its uppermost position. The cycle of operation is therefore automatically repeated until the movement of the push rod becoming shorter and shorter, finally ceases

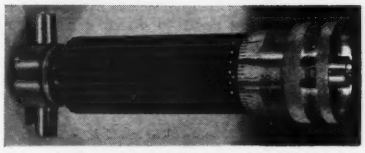


Fig. 1. Dobbie McInnes peak pressure indicator.

altogether in a few cycles and rotation of the thimble also stops. The last increment of movement is barely perceptible and it is stated that any inertia effect entirely disappears. This accounts for the accuracy of the indicator and the complete absence of "over shooting". It also permits a

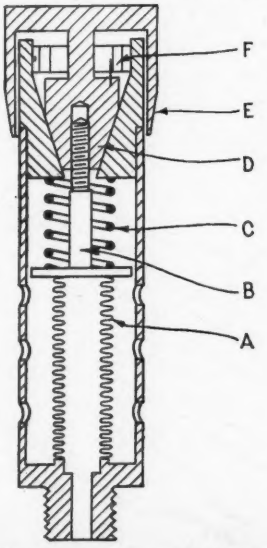


Fig. 2. Diagram showing working principle.

true reproduction of working conditions to be obtained when calibration is performed on an ordinary hydraulic type pressure gauge tester.

When the peak pressure is reached, the push rod is locked in its uppermost position. To re-set for immediate repeat readings, the thimble is simply twisted in a clockwise direction. Otherwise the indicator is not touched by hand, and, since operation is automatic, personal errors of judgment are avoided. The bellows unit, which is of non-corrosive material, will withstand temperatures up to 500 deg. C, and test pressures of more than double the maximum cylinder pressures of internal combustion engines.

The indicator was primarily evolved to provide a reasonably priced but reliable instrument for medium and high-speed internal combustion engines. It shows at once whether there is unequal loading of the cylinders through leaking piston rings and valves, for example, and in the case of diesel engines through unequal metering of fuel, choked atomizers or other causes.

The following is a summary of the advantages claimed for this instrument:—(a) It is automatic. (b) Cleaning is only required at infrequent intervals. (c) Accuracy does not depend on personal judgment. (d) Static calibration is true. (e) It is very rapid in operation. (f) There are no pistons, valves or mechanisms such as pawls, ratchets or gears. (g) Friction is negligible. (h) Cost is low.

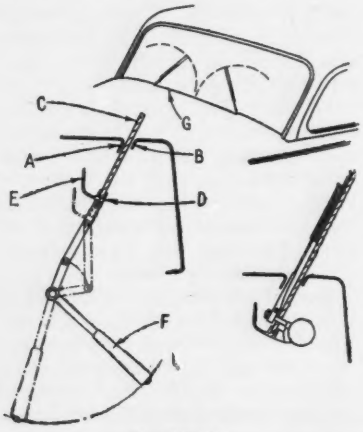
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Retractable Windscreen

TO minimize obstruction of vision and obtain a clearly defined line at the upper face of the bonnet, an unframed windscreen panel is mounted inside the body structure and slidable in guideways on the front pillars. The top of the bonnet terminates in a downwardly extending flange A, and the upper run of the body component forming the instrument panel is similarly flanged at B. These two flanges are spaced apart to provide a slot through which windscreen panel C extends. The lower edge of the windscreen is mounted in a support D, located inside the bonnet, which is provided with a transverse gutter E to collect rain which may flow down the outer face of the windscreen. Opposite ends of this gutter are furnished with flexible rubber discharge conduits.

A toggle linkage may be provided for raising and lowering the windscreen. In the example shown, this is operated by a lever F below the instrument panel. One or more wiper motors may be mounted on the windscreen support D with their shafts extended through holes in the panel. The rear edge of the bonnet top will be

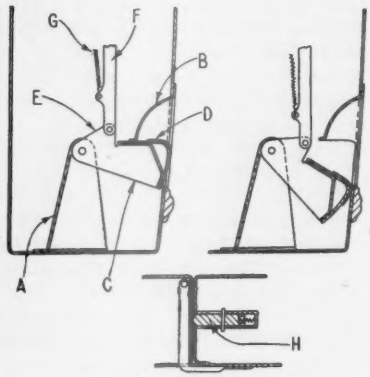


No. 632116

recessed at G to provide clearance for the wiper blades, which lie below the bonnet top when not in use. *Patent No.* 632116. *Briggs Motor Bodies, Ltd., and J. Tilbury.*

Foot-Operated Door Latch

IN order to dispense with a handle on the exterior of a car door, a conventional latching device is actuated by a pedal housed in the base of the door. A sheet metal bracket A, comprising side walls and an inner wall, is mounted on the lower run of the door. Above it, and attached to the outer panel, is an arch B bordering a recess formed in the outer face of the door. Pivoting on a horizontal spindle carried by bracket A is the pedal C to which is secured a rubber tread D. The side walls of this pedal are each formed with a lug E, to one of which is pivoted a rod F to operate the door latch. Two lugs are provided to make the pedal suitable for operation of either right- or left-hand latching arrangements. A tension spring G serves to return the lever to its inoperative position.



No. 632122

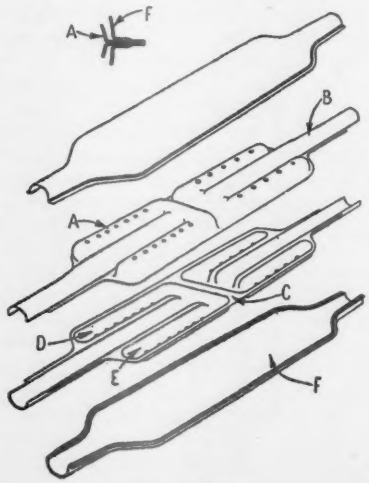
To release the door, a toe is inserted in the arch and the pedal depressed to withdraw the latch. The door opens sufficiently to allow a hand to be passed behind the edge to complete the opening movement, as a spring-pressed plunger H is fitted in the door to thrust against the jamb. Alternatively, some other spring device may be used to effect a partial or full opening of the door.

The specification also suggests that the pedal could be fitted in the jamb instead of in the door. *Patent No.* 632122. *Austin Motor Co., Ltd., and L. P. Lord.*

Silencer Construction

THIS invention relates to silencers of the type constituted by spaced inner and outer casings. The inner casing is divided into inlet and outlet compartments and the transfer of gas from one to the other is by way of the spaces between the two casings.

Each casing is formed of two similar pressings having outwardly directed mating flanges along a longitudinal plane of division. For the inner casing, the pressing A is of an approximate rectangular form with integral semi-circular channel extensions B at both ends. These channels



No. 632013

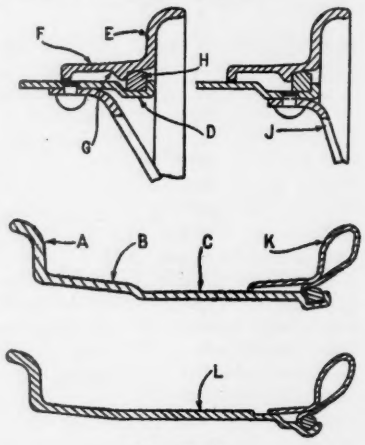
are continued longitudinally up to the central rib C dividing the inlet side from the outlet side. From the central rib the channel divides into two return channels D and E which are furnished with a series of perforations to establish communication with the space between casings.

Outer casing pressings F fit over the inner casing leaving a space for the passage of gases on each side of the plane of division. The outer casing has flanges joggled to receive the flanges of the inner casing and channel extensions to embrace those of the inner casing.

To assemble, the flanges of the inner pressings are independently spot-welded, this inner assembly is tacked by spot welds to one of the outer pressings and finally the outer casing flanges are continuously welded. *Patent No.* 632013. *British Light Steel Pressings Ltd., and E. S. White.*

Detachable Flange Rim

AIMED to facilitate mounting and demounting of the tyre, this rim has an endless detachable flange retained by a



No. 631101

split clamping ring. The base member is formed with tyre retaining flange A, a bead seat B, a shallow well C and a gutter D. Telescoping over base portions D and C, the detachable flange E embodies a bead seat F and on its inner periphery a shoulder G to engage clamping ring H. Outboard of the shoulder the profile is flared at an angle of approximately 12 deg. to engage a complementary surface on the outer periphery of the substantially rectangular section clamping ring. The ring profile is symmetrically tapered so that the ring is reversible. When the tyre is inflated the relatively rigid detachable flange exerts a wedging stress through the clamping ring which compels radial flexure of the gutter.

In a modification for a wheel in which the body J is dished to a less extent, the flexing action is transferred also to the peripheral portion of the wheel body.

Other constructions use a detachable flange K formed of one piece of sheet metal. For the modified base L a sheet metal detachable flange is retained by a clamping ring of circular cross section. *Patent No.* 631101. *Kelsey-Hayes Wheel Co. (U.S.A.).*